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DIGITAL FLIGHT CONTROL ACTUATION SYSTEM STUDY

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FINAL REPORT

BY R. ROSSING AND R. HUPP

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HAMPTON, VIRGINIA 23665

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FOREWORD

This report documents work performed by the Columbus Aircraft Division of Rockwell International Corporation at Columbus, Ohio, for NASA-Langley Research Center, Hampton, Virginia, under Contract NAS1-12718. Technical direction was administered by Mr. Robert Averill, Technical Representative, Mechanical Systems Section, System Development Branch, Systems Engineering Division, NASA-Langley Research Center (Mail Stop 315).

This study program was conducted to determine the most suitable approach for development of an aircraft flight control actuation system for use in an advanced redundant, all digital fly-by-wire flight control system. The study effort consists of two parts; (1) a state-of-the-art review and comparative analysis, and (2) a conceptual design study. The first part of this study is documented in two sections of this report; Section 2, State-of-the-Art Survey, and Section 3, Comparative Analysis. The results of the second part of this study are documented in Section 4, Conceptual Design Study. The following engineers assisted in this study and made significant contributions; J. Berry, L. Grieszmer, R. Haning, R. Martin, and W. Mathena.

Two system of units are utilized throughout this report, SI and English. The SI units are presented first followed by the customary units in parentheses. The customary English system was used for principal measurements and calculations and then converted to SI units.

ACKNOWLEDGEMENT

Appreciation is extended to the many individuals who provided helpful suggestions and constructive criticisms. Specifically Mr. R. Averill of NASA-Langley and Messrs. S. Burtner, C. Knox, and Dr. F. Bellar, Jr. of the Columbus Aircraft Division of Rockwell International Corporation.

ABSTRACT

The objectives of this program were to review the state-of-the-art of flight control actuators and feedback sensors suitable for use in a redundant digital flight control system and to select the most appropriate design approach for an advanced digital flight control actuation system for development and use in the second phase of NASA's digital fly-by-wire experimental program. A survey was conducted to accumulate data and ideas on fly-by-wire power actuators and control devices, transducers, and the control to actuation system interface. Results of this survey are contained in Section 2, State-of-the-Art Survey. The results of the survey were reviewed and analyzed and are presented in Section 3, Comparative Analysis. Four design approaches, embodying the best features for this program, were selected and studied in detail. A trade study was conducted on these four configurations, all of which are considered acceptable for the program. Three approaches were carried through the conceptual design phase. The fourth approach was recommended for separate study. The conceptual design study, documented in Section 4, resulted in the selection of the PM torque motor direct drive as the optimum approach to satisfy the requirements of the NASA digital fly-by-wire program. The conceptual design study did not uncover any reason which would preclude using this approach on the Phase II aircraft. Further, the PM torque motor direct drive approach is compatible with concurrent and independent development efforts on the computer system and the control law mechanizations.

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LIST OF ABBREVIATIONS AND ACRONYMS

ac, AC	alternating current
Act.	actuator
AMP	amplifier
Assy	assembly
AFTI	advanced fighter technology integrator
BCS	backup control system
BIT	built-in test
BITE	built-in test equipment
BRK	break
CCV	control configured vechile
CH	channel
CMOS	complimentary metal oxide semiconductor
$C_1, C_2,$ ---	elements in reliability diagram
D/A, D\A	digital to analog
dc, DC	direct current
D/D	digital to digital
DEMOD	demodulator
DET	detector
Dia.	diameter
DIGIFLIC	digital flight control
DIFFER.	differential gearbox
DIP	dual inline package
ELEC.	electrical
E/H	electro-hydraulic
EM	electro-mechanical
FBW	fly-by-wire
FPFH	failures per flight hour
GEN.	generator
gpm	gallons per minute
hp	horsepower
HYD	hydraulic

IAP	integrated actuator package
IC	integrated circuit
IMU	inertial measurement unit
I/O	input to output unit
LED	light emitting diode
LIM	limiter
LSB	least significant bit
LSI	large scale integration
LPFH	aircraft losses per flight hour
LVDT	linear variable differential transformer
MAX	maximum
MOS	metal oxide semiconductor
MSB	most significant bit
MTBF	mean time between failure
NMOS	N-type metal oxide semiconductor
No.	number
PC-1, PC-2	hydraulic system nomenclature
PM	permanent magnet
PMOS	P-type metal oxide semiconductor
pps	pusles per second
RVDT	rotary variable differential transformer
SAS	stability augmentation system
SCR	silicon control rectifier
TD	time delay
T.M.	torque motor
V	volts
VR	variable reluctance
V/STOL	vertical or short take-off and land
W	watts

MATHEMATICAL SIGNS AND SYMBOLS

A_1, A_2, A_3, A_4	elements in reliability diagram
A_p, A	primary actuator area
B_L	primary actuator viscous friction coefficient
B_V	valve viscous friction coefficient
$B_1, B_2, B_3, \text{---}\ B_8$	elements in reliability diagram
C_{tp}	actuator leakage coefficient
$C_1, C_2, \text{---}$	elements in reliability diagram
F	force
F_S	stiction force
g	gravitational acceleration
H	position feedback gain
$I_{avg.}$	average current
K_A	current amplifier gain
K_C	valve pressure/flow coefficient
K_F, H	actuator position feedback gain
K_g	gear ratio
K_I	current feedback gain
K_L	load spring rate
K_P	valve pressure gain
K_q, K_Q	value flow gain
K_T	torque coefficient
K_X	spring coefficient
K_V	value spring rate
K_ν	velocity constant
K_β	actuator compressibility coefficient
M_L	load mass

MATHEMATICAL SIGNS AND SYMBOLS (Continued)

M_V	valve mass
P	load pressure
$Q, Q_A, Q_B, ---Q_H$	probability of failure
Q_P	probability of pump failure
Q_D	probability of hydraulic distribution system failure
Q_{ACT}	actuation system failure probability
Q_{COMP}	computer system failure probability
Q_{ELEC}	electrical system failure probability
Q_{SENS}	sensor system failure probability
Q_{STICK}	control stick assembly failure probability
$Q_{SURFACE}$	control surface failure probability
R	reliability number
S	Laplace operator
T	time constant
X_{MAX}	maximum linear rate
X_V	valve position
X_P	piston position
q	flow
Δi	torque motor current
Δp	differential pressure
ϵ	servo error signal
ζ	damping ratio
λ	failure rate
σ	current amplifier fdbk signal
τ	torque
χ_c	coulomb driction coefficient
χ_s	stiction friction coefficient
ω	natural frequency

SUMMARY

The method used to select the most appropriate design approach for an advanced digital flight control actuation system for development and use in the second phase of NASA's digital fly-by-wire experimental program was divided into two parts. The first part consisted of a state-of-the-art survey followed by a comparative analysis of the survey results. The second part was a conceptual design study to select the optimum design approach.

The goal of the state-of-the-art survey of the study was to conduct and document an in depth survey and review of current state-of-the-art advanced flight control actuation and feedback system design concepts which could meet the requirements of Digital Fly-by-Wire Primary Control Systems. The variety of concepts considered included hydraulic, electric, pneumatic, mechanical; flown, tested, proposed; rotary, linear and integrated. Approximately 210 reports, papers, articles, etc., were reviewed in the literature search. A letter survey of approximately 111 manufacturers was conducted. Aprroximately 20 plant visits, both in-house and out-of-house, rounded out the quest for data.

The results of the search indicate that the majority of the systems were analog/hydraulic with little developmental effort on direct digital actuation. The prevailing sentiment indicates that electronic digital/analog and analog/digital control is not a problem and that servo loops can be closed digitally. Development work is being conducted on electro/mechanical (E/M) actuation systems primarily because of the space shuttle requirements although no operational primary flight E/M systems were uncovered. The activity in pneumatic or mechanical actuation systems of note has been in engine controls and with the Harrier aircraft. A brief description of this and other relevant concepts are contained in Section 2. The concepts presented are not meant to be all inclusive, but do represent a cross-section of current thinking on primary flight control actuation systems.

A large number of systems or concepts were reviewed, and are included, which could be used in digital fly-by-wire primary control systems. Much work has been completed and is continuing in the development of actuation systems which would be suitable for the NASA program. These ranged from an abundance of dual tandem power actuators and secondary actuators to one of a kind integrated, digital, and direct drive actuators.

SUMMARY (Continued)

The numerical control field is providing the majority of actuation systems capable of accepting direct digital inputs. The usual form is an electrical stepper motor controlling a hydraulic drive motor. A few developmental actuators using a stepper motor to control the conventional flight control spool/sleeve valve actuator were identified.

The predominant power actuator both in use and in development is the linear actuator controlled by a spool/sleeve valve regardless of the method used to control the spool.

The Linear Variable Differential Transformer (LVDT) is by far the most prevalent feedback sensor in use. The LVDT performs well and the basic need to develop a digital sensor was not identified. Digital encoders of various types are available and were identified in the survey.

A prevailing trend was noted toward mechanizing the logic functions in the electronics rather than in the actuator. Developmental work is being performed with hydraulic logic but most companies covered in the survey preferred to accomplish the task with electronics.

Developmental work on integrated actuator packages, both E/M and electrically driven hydraulic pumps, together with remote, but not integrated, power systems were identified in the survey. The servo pump was developed to reduce power loss in these integrated packages by generating hydraulic power only on demand. This is practical due to the control power requirement being high instantaneously but relatively low on an average basis. One scheme using a flywheel in conjunction with an electric motor to accomplish this leveling of the power required was identified.

A comparative analysis of the fundamentally different approaches to digital flight control actuation systems identified in the state-of-the-art survey was then made to continue towards the selection of the most appropriate approach for development in the NASA digital fly-by-wire experimental program. The method used in this analysis consisted of the following steps: (1) performance requirements were projected for next generation aircraft, (2) a general review and evaluation of components and techniques found appropriate for digital flight control actuation systems was made, (3) four approaches incorporating the most promising and fundamentally different principals were selected for more detailed consideration, and (4) a trade-off comparison was made between the selected approaches.

SUMMARY (Continued)

Actuation system requirements were established for two types of aircraft, high performance and commercial transports, by reviewing present aircraft needs and projecting future needs. General requirements for force, rate, stroke, stiffness, bandwidth, resolution, backlash and reliability were established. These requirements provide a realistic base for selecting, rejecting and comparing the various approaches considered.

An evaluation of components and devices identified in the survey was then conducted to aid in selection of the most promising approaches. This effort encompassed power media, control, interface devices, sensors, and redundancy mechanizations. Four candidate systems were configured and a trade-off study was conducted which considered loss rate reliability, operational reliability, maintainability, design simplicity, performance, cost, weight, and limiting factors. The four candidate systems were; (1) PM Torque Motor Direct Drive, (2) Stepper Motor Direct Drive, (3) Secondary Actuator, and (4) Electro Mechanical.

The trade-off study indicated a preference for one approach, however, all four of the concepts were acceptable. The three hydraulic approaches were recommended for further study in the conceptual design phase. The fourth approach, EM, was recommended for investigation under an additional concurrent effort.

A secondary actuator approach was included as a baseline system for comparison because the equipment, knowledge and test data currently available for this type of system are the most highly developed, and as such represent the lowest risk approach. It could be implemented at the present time.

A system configuration using a torque motor to directly drive the power spool of the surface actuator was recommended as one of the system configurations to be carried through the conceptual design phase. This is felt to represent the simplest, most direct and foolproof approach to the present program. The torque motor would be constructed from advanced high strength magnetic materials and would be mechanically coupled to the power spool. Feedback would be provided by a conventional linear variable differential transformer (LVDT).

The stepper motor direct drive configuration consists of dual electrical stepper motors, a differential summing mechanism, a mechanical feedback assembly, and a power spool/sleeve valve. The dual stepper motors and mechanical feedback signal mechanically sum in the differential mechanism, the output of which drives the power spool/sleeve valve. The power valve may be a two stage hydraulic helix design developed for NASA-MSC. The rotary mechanical feedback signal is generated by a helical screw within the actuator rod.

SUMMARY (Continued)

Electro-mechanical actuators in the past have generally been relegated to secondary controls primarily due to performance, weight and reliability considerations. Electrical technology has advanced considerably, especially in solid state controllers and also in electric motor design. It now appears, adequate response, weight, and reliability for some primary control applications can be obtained from electric motors in closed loop actuation systems. Electrical power generation, distribution, and actuation (i.e. power-by-wire) offers advantages in power system fault detection and isolation, reduced fire hazards, and possibly in ease of maintenance and repair. Considerable interest in this area was generated by Space Shuttle work. The ultimate limitation and role of EM in flight controls could not be established in this limited study effort. It is believed however that EM actuation is worthy of further investigation. This can be best pursued by specific design, development and testing of flight worthy hardware.

The three selected approaches; PM Torque Motor Direct Drive, Stepper Motor Direct Drive, and Secondary Actuator, were carried through the conceptual design study. The initial task was to present a preliminary trade-off of the factors involved in the application of the three configurations to the proposed Phase II aircraft. A preliminary design was then undertaken to identify any problem areas which would preclude development of any of the three selected configurations for use in the Phase II aircraft.

The preliminary design did not uncover any problem area which would restrict in any manner using any of the three configurations on the test aircraft. All three designs are flexible and offer many acceptable variations in the actual design for installation in the test aircraft. Program considerations of time, cost, and the performance required for the investigation of control laws will dictate the ultimate actuation system design. It was also verified that the three selected configurations would be compatible with concurrent and independent development effort associated with the computer system and the control law mechanization.

The results of the trade-off of the three approaches indicated the PM Torque Motor Direct Drive configuration would best satisfy the requirements of the NASA digital fly-by-wire program. The PM Torque Motor Direct Drive design approach was selected primarily on the basis of performance, simplicity, reliability, and compatibility with future digital fly-by-wire requirements. A recommendation is made that this configuration be developed for the initial flight program utilizing a multi-channel digital fly-by-wire system.

SUMMARY (Continued)

It should be noted that this review and analysis was restricted to the actuation system. The power generation source was not analyzed. Work is being performed on high voltage and frequency electrical power systems; very high pressure hydraulic systems, and on centralized, localized, and integrated hydraulic systems. The impact and potential of the various power generation schemes on the actuation system should be the subject of further study.

SECTION 1

INTRODUCTION

NASA has initiated a program to develop the technology required to implement advanced, reliable, digital fly-by-wire flight control systems which will increase operational capability and performance of future aircraft. The overall digital fly-by-wire experimental program will be accomplished in two phases designated as Phase I and Phase II. The Phase I effort is being directed toward the development and flight demonstration of a single-channel, three-axis, digital fly-by-wire primary flight control system. The Phase II program will develop and demonstrate a redundant multi-channel, all-digital fly-by-wire flight control system with advanced control capability. Phase II will provide and verify an aircraft compatible design base for digital fly-by-wire flight control systems and will permit investigation of advanced control law concepts.

An important aspect of the digital fly-by-wire experimental program is development of an advanced flight control actuation system compatible with digital flight control computer requirements. Advanced applications necessitate a flight control actuation system have improved performance with appropriate fast response, accuracy, and reliability over a wide range of environmental conditions.

There have been numerous fly-by-wire investigation programs over the past several years; however, little current emphasis has been placed on the development of actuation systems expresseIy suitable for use with redundant, all-digital, fly-by-wire control systems for aircraft.

The purpose of the research study described herein was to review and advance the state-of-the-art of flight control actuators and feedback sensors suitable for use in multi-channel digital flight control systems and to determine the most suitable approach for the development of an aircraft flight control actuation system for use in the Phase II aircraft. The study is divided into to major parts as delineated in Figure 1.0-1. The first part of the study consisted of a review and summary of the current state-of-the-art of actuation devices suitable for use in a digital flight control actuation system. Section 2 covers the state-of-the-art survey. Appendices A, B, C, and D include supporting detailed information with regard to the state-of-the-art survey. This information was used in a comparative analysis to determine the most promising concepts for an advanced, digital, flight control actuation system design. This analysis considered applicable current system concepts,modifying such systems as necessary by assuming the use of interface equipment such as D/A converters and servo-amplifiers required for digital input compatibility. Promising digital actuation devices not previously used in an aircraft application were considered with appropriate modifications to make them suitable for the proposed Phase II flight application. The results of the comparative analysis are contained in Section 3. Three approaches were selected from the comparative analysis for more detailed study and preliminary design in the second part of the study.

The second part of the study consisted of a conceputal design study to determine the factors involved in the application of the three proposed advanced digital actuator design concepts to an experimental aircraft. A system trade-off study was conducted to consider the factors involved in application of the proposed design concepts to pitch, roll, and yaw control of the Phase II flight aircraft. This trade-off considered the use of the existing aircraft power control hydraulic systems, redundancy provisions to assure flight control actuation system reliability, performance, and interface requirements. A preliminary design was then conducted on the three concepts. This preliminary design included consideration of packaging, sizing, and installation factors, with the intent to identify any problem areas which would preclude development of the concept for use in the Phase II aircraft. The data from the trade-off and preliminary design was then accumulated to select a single design. The PM torque motor direct drive concept was selected as the design concept which best satisfies the requirements of the proposed application and was recommended for further development. A mathematical model of the PM torque motor direct drive system was then developed for real-time simulation of the design on a hybrid computer. This conceptual design effort is contained in Section 4. The contractors recommendations are contained in Section 5.

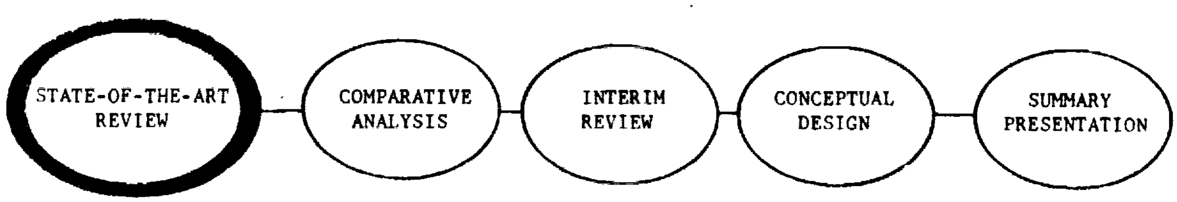


FIGURE 1.0-1 DIGITAL FLIGHT CONTROL ACTUATION SYSTEM PROGRAM

SECTION 2

STATE-OF-THE-ART SURVEY

2.1 INTRODUCTION

NASA has initiated a program to develop the technology required for implementation of advanced, reliable, digital fly-by-wire primary flight control systems which will provide greater operational capability and increased performance of future aircraft.

The purpose of this portion of the program was to conduct and document an in-depth survey and review of current state-of-the-art actuation concepts applicable to digital fly-by-wire primary flight control systems. The survey was not restricted to direct digital actuation systems. Analog actuation systems which could be adapted to digital control were included. Analog actuation systems, in fact, constitute the majority of contemporary systems. All likely information sources were investigated to uncover data on past, present, and planned projects. Information concerning the literature search, letter survey, and visitations can be found in paragraphs 2.1.1 through 2.1.3. The approach taken in conducting the state-of-the-art survey is depicted in Figure 2.1-1.

The results of the survey are compiled in paragraph 2.2 illustrating various types of components, control signals, methods of transmitting signals, actuator configurations, degrees of redundancy, and failure detection techniques. These concepts were uncovered during the survey of the established sources in the field. There was no attempt to analytically challenge the results uncovered. The information was taken from data accumulated for this study and acknowledgement is given to the contributor where possible. The concepts are categorized in Figures 2.2-1 through 2.2-3 and Figures A-1 and A-2 of Appendix A, by subdividing into functional groups; i.e., actuation systems, components, etc.

A summary discussion of this portion of the study including prominent trends indicated by the results is contained in paragraph 2.1.4. Detailed bibliography of the literature reviewed searched, a listing of the companies surveyed by letter, and the company and agencies visits are presented in Apprneices B, C, and D.

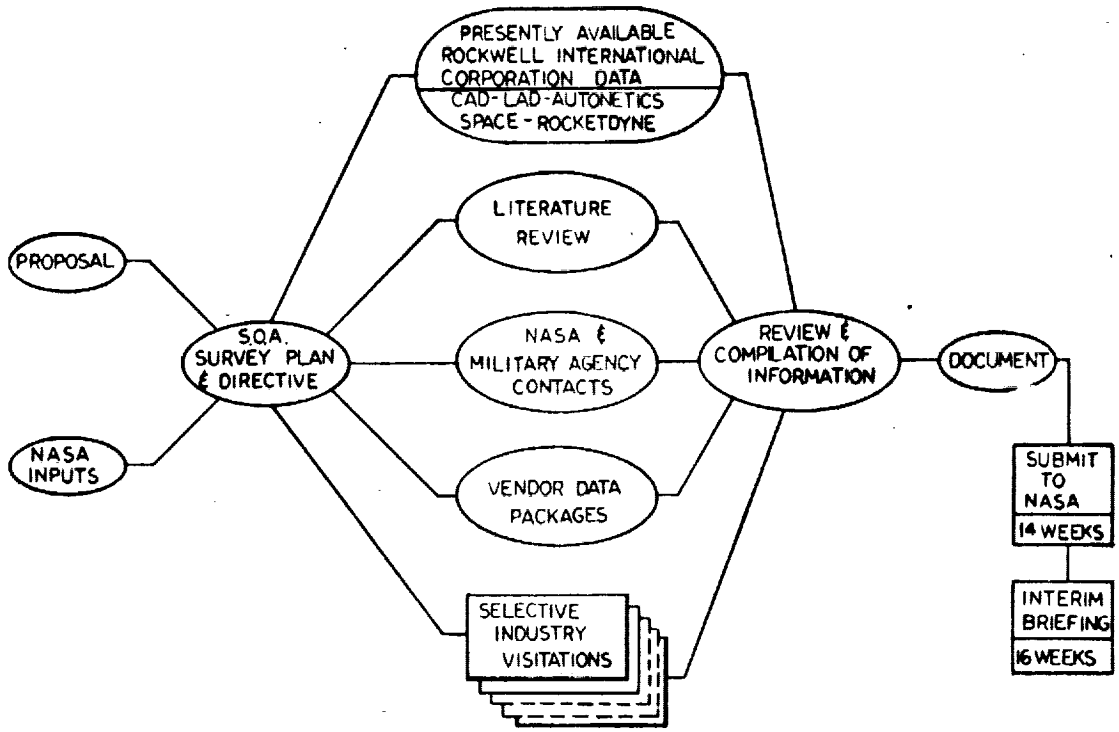


FIGURE 2.1-1 DIGITAL FLIGHT CONTROL ACTUATION SYSTEM STATE-OF-THE-ART

2.1.1 LITERATURE SEARCH

Because of the relative newness of the art of fly-by-wire design, no single source of data is available; therefore, all likely information sources were investigated during the study period to uncover data on past, present, and planned systems. To cover as many sources of related work as possible, a literature search, a letter survey, and a company visitation program were carried out. The literature search included an Abstract Bibliography Request from the Defense Documentation Center, a NASA literature search on digital flight control actuators, a retrospective search of the Mechanized Information Center, Ohio State University Libraries, a Rockwell International TIPS Computer Search on actuating devices for digital flight control systems, and hand search of the Technical Abstract Bulletin, Science and Technical Aerospace Reports, and International Aerospace Abstracts. The results of the search are found in, Appendix B, the Literature Search Bibliography, of this report. The Bibliography is not intended to be all inclusive but does represent an excellent cross section of available literature on the subject of actuation system and fly-by-wire systems. There is no shortage of literature on the subject.

2.1.2 LETTER SURVEY

A letter survey into actuators and sensors suitable for use in digital fly-by-wire flight control system was conducted. The survey letter was sent to the potential suppliers, manufacturers, and equipment users listed in Appendix C requesting their participation in the survey by supplying information regarding actuation systems, components and/or experience which may have application in a multi-channel digital fly-by-wire system. The survey letters were sent to one hundred and eleven (111) companies with approximately one-third of the companies/agencies responding. As with most letter surveys, the results were mixed. Very few in-depth results were obtained directly from the letter survey without some personal follow-up. The letter survey was beneficial in opening leads to work that might have been overlooked had the survey not been conducted.

2.1.3 PLANT VISITATION

Since the advanced work and thinking related to fly-by-wire design has not yet found its way into the literature, plant visits were made to various airframe and actuator companies to determine their attitudes, past work, or plans (if any) involving fly-by-wire control. Several companies personally delivered their information and contributed their personal knowledge and experience on the subject. A listing of companies personally contacted is contained in Appendix D.

The person to person discussion was by far the most rewarding and informative method of obtaining information attempted in this survey. There does not appear to be a suitable alternate to face-to-face discussion. Many thanks to the numerous people who contributed their time, talent, and energy to this survey.

2.2 COMPILATION

A broad variety of concepts were investigated with the goal of identifying components or design concepts which might be suitable for aircraft application.

Detail examples of the concepts which illustrate the various types of components, controls, actuator configurations, degrees of redundancy, and fail detection techniques are contained in Appendix A. These concepts were uncovered during the survey of the established sources in the field. There was no attempt to develop new concepts or to analytically challenge the results uncovered. The information was taken from the data accumulated for this study and acknowledgement is given to the contributor where possible.

Each concept is assigned a sequencial design concept number which is explained by the concept numbering key preceding the eighty-three (83) concepts presented in Appendix A. Each concept consists of a brief description of operation, a block diagram, and/or a hydraulic schematic if available. In addition to identifying the individual concepts, the design concept number provides the means to correlate the information contained therein to the parent literature from which it was taken. The last three (3) digits of the concept number represent the literature find number listed in the literature search bibliography located in Appendix B of this document.

The concepts presented are not meant to be all inclusive, but do represent a cross section of current thinking on primary flight control actuation systems. The concepts are summarized, in matrix form, in Figure A-2 of Appendix A which is subdivided into function groups; digital hydraulic actuation, actuation, components, etc.

Figures 2.2-1, 2.2-2, and 2.2-3 are flow charts showing three major areas considered in this study. The flow charts are further broken down within each of the major areas to eventually show the applicable concepts listed in Appendix A. The three major areas are; Digital Hydraulic Actuation (Actuators), Redundant Actuation (primary actuator systems), and Components, Systems, and Integrated Actuator Packages applicable to a fly-by-wire concept. An explanation of each flow chart is given in the following paragraphs.

2.2.1 DIGITAL HYDRAULIC ACTUATION

Digital hydraulic actuation concepts can be classified in accordance with Figure 2.2-1. Basically they fall within two groups, quantized rate or quantized position. Typical quantized rate type concepts found were flow summation i.e., flow from on-off valves are summed to produce total actuation rate, stepper motor driven flow control valves, and various approaches to a digital driving of analog type valves. The quantized position concepts reviewed operate on a digitized volume or digitized step principle. Digitized volume devices move incrementally by injecting a fixed quantity of fluid into the actuator. Digitized stepping devices utilize some form of feedback to close the valve after it has moved the commanded increment. Examples of each of the concepts are presented in Appendix A with concept index numbers listed in Figure 2.2-1.

2.2.2 REDUNDANT ACTUATION

Considerable effort has been expended over the past decade in the study and development of redundant actuators and actuation systems as attested by the volume of literature available. The evolution began from simple limited authority fail safe parallel or series actuators for electronic flight control commands (SAS) and has progressed through one and two fail operational full authority fly-by-wire systems. The Redundant Actuation concept can be classified in accordance with Figure 2.2-2. The first level of breakdown is the basic approach to redundancy; active, standby, or some combination of two, hybrid redundancy. Most of the work to date has been in either the active or standby categories. The first level breakdown can be further divided into groups defining the power level to which the redundancy is carried; control power level or output power level. For example, a standby system which utilizes multiple secondary actuators to control the surface actuator would be listed under the control power category. The control power category can be further divided by whether the redundant channels combine at a secondary actuator or at a direct drive valve. As shown in Figure 2.2-2, a fourth and fifth level categorization is made in the active redundancy branch depending upon whether the system utilizes force, rate, or position summation and further by whether the secondary actuator is hydraulic or electro-mechanical. Examples of each category can be found in Appendix A from the concepts listed. As indicated by the number of examples, the active redundant, secondary, force summed, hydraulic actuator and the standby, secondary actuator approaches have been selected in the majority of the past development efforts.

2.2.3 COMPONENTS, SYSTEMS, AND INTEGRATED ACTUATOR PACKAGES

A representative collection of Components, Systems, and Integrated Actuator Packages (IAPS), found in the literature search and survey which are applicable to digital control systems or primary flight control systems are included in Appendix A. Figure 2.2-3 lists by index number those concepts presented.

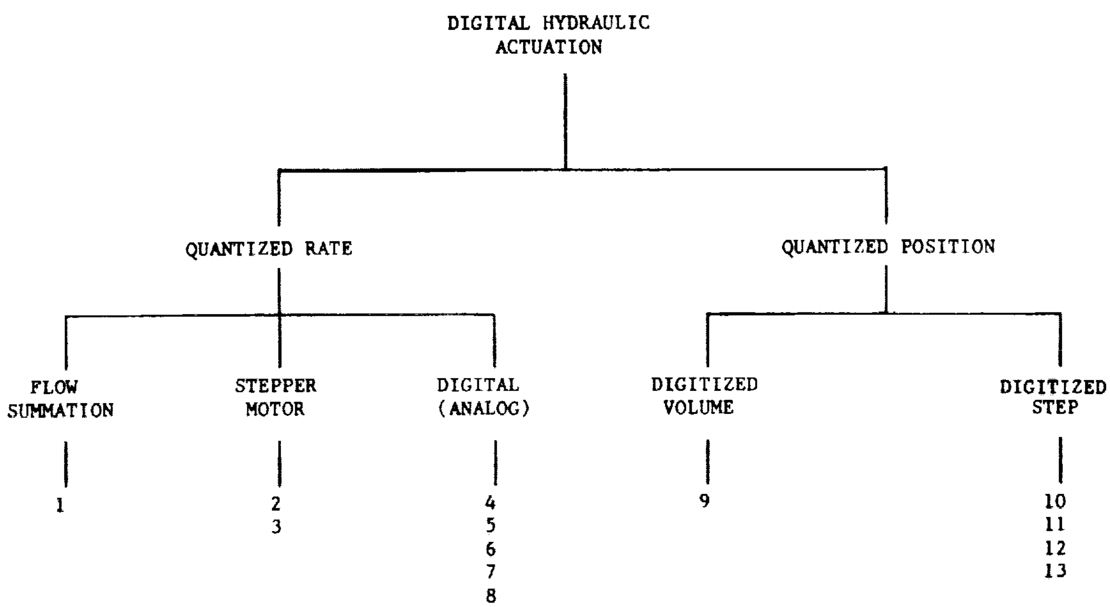


FIGURE 2.2-1 FLOW DIAGRAM-DIGITAL HYDRAULIC ACTUATION

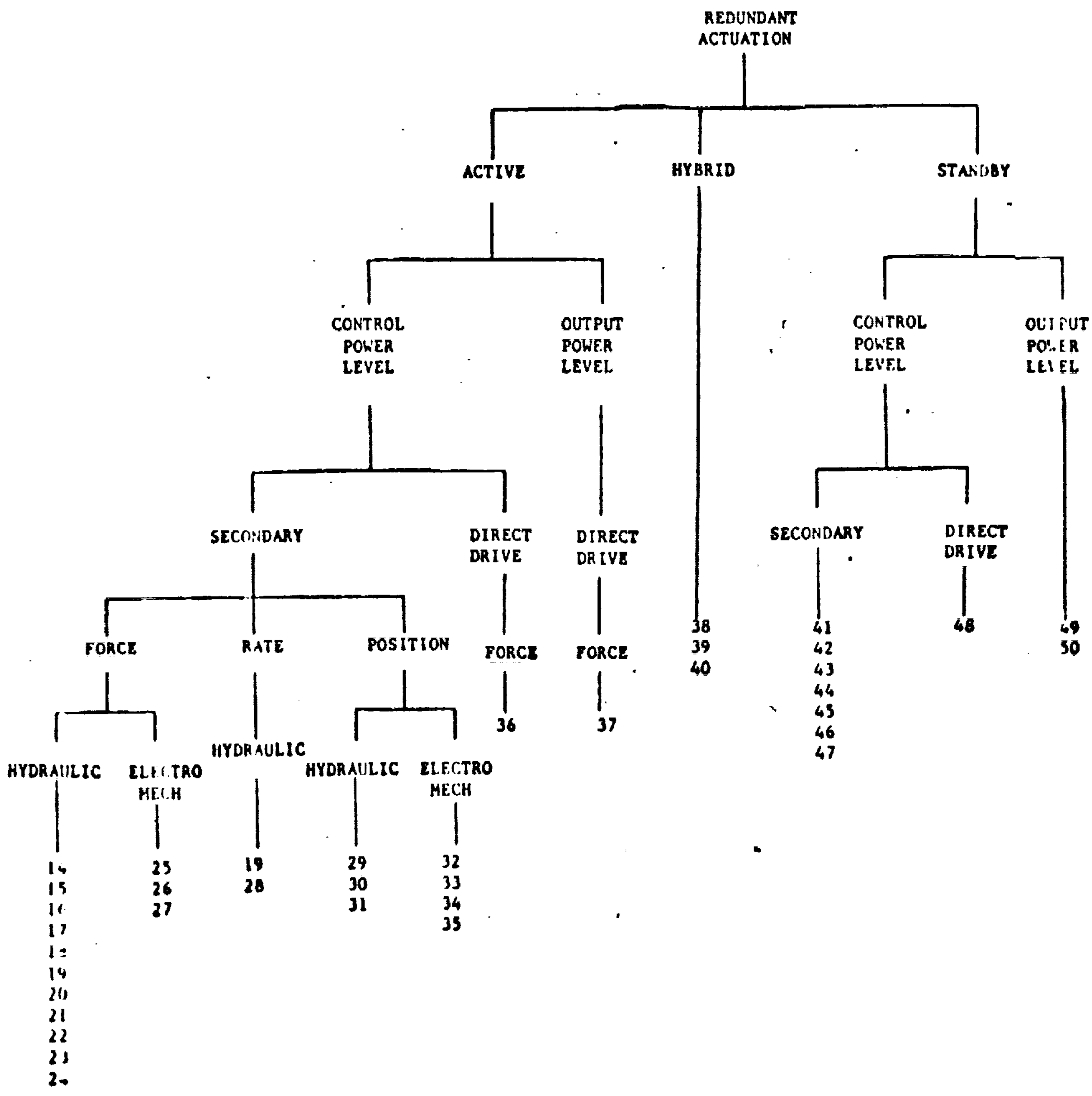


FIGURE 2.2-2 FLOW DIAGRAM-REDUNDANT ACTUATION

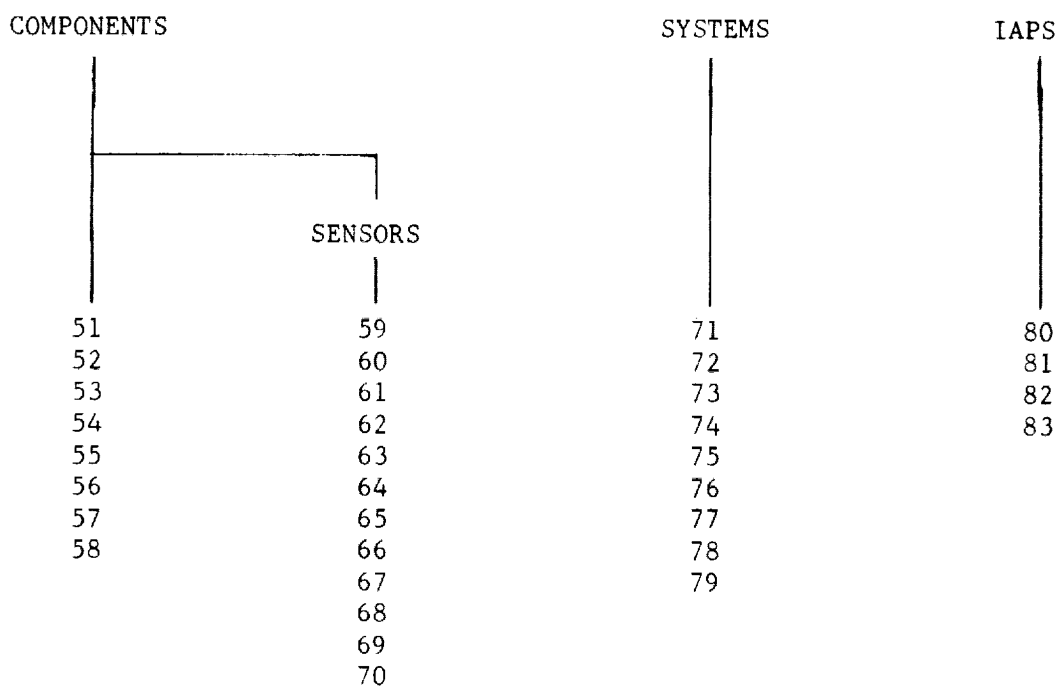


FIGURE 2.2-3 FLOW DIAGRAM-COMPONENTS, SYSTEMS AND IAPS

2.3 STATE OF THE ART SUMMARY

The objective of the initial phase of this study was to conduct a survey of the state-of-the-art for Digital Flight Control Actuation Systems. The survey was divided into four basic activities: (1) a literature search to uncover data on past, present and planned systems; (2) a letter survey requesting the participation of potential suppliers, manufacturers and equipment users to supply any information regarding actuation systems or components for application in multi-channel digital fly-by-wire systems; (3) plant visits to determine company opinions, past work, or plans involving fly-by-wire control; and (4) a compilation of this accumulated data containing the systems and components considered along with respective sketches, diagrams and schematics when available.

Over two hundred ten (210) pieces of literature were reviewed, approximately one hundred eleven (111) companies were surveyed by letter and eighteen (18) companies visited or were visited to discuss and exchange views on the subject. The areas of interest to this survey included concept classification (direct drive, digital, analog), Identification (association with current programs like F-14, F-15, F-111, Space Shuttle, Harrier, 680J, AFTI), Redundancy (fail operate, fail safe, fail to null), Control (Jet Pipe, Flapper, Clutch, Solenoid), Power (hydraulic, pneumatic, fluidic, mechanical), Power Source (aircraft engine, electric motor, battery), Actuators (dual or triple tandem and dual, triple and quadruplex parallel), Feedback Sensors (LVDT, RVDT, encoders), Monitoring (pressure, hydraulic, position, electrical), Status (conceptual, experimental, breadboard, prototype, production), and any comments concerning the concept itself.

Some of the prominent trends resulting from the survey are as follows:

A) Analog electro-hydraulic configurations dominate the review. Electro-mechanical actuation systems are being investigated primarily as a result of space shuttle program interest. No development work on pneumatic or mechanical power systems for primary flight controls was uncovered. The Harrier aircraft does use a pneumatic actuation system for thrust control, and pneumatics is used extensively for engine controls.

B) The advantage of electro-mechanical configurations needs to be proven by further testing and application.

C) Compromises must be made in redundant systems because of the added complexity, maintenance and provisioning problems. Much disenchantment with the proliferation of redundancy to meet fail safe reliability requirements was noted.

2.3 STATE OF THE ART SUMMARY (Continued)

D) Digital concepts, D/A or A/D conversion, closing servo loops digitally, and driving analog components digitally is not a problem with current experience and technology.

E) Higher pressures are reducing actuator sizes. Integrated and micro electronics are shrinking electronic package sizes.

F) Force summing, velocity summing, and position summing designs are available for use in redundant systems. Each has pro's and con's.

G) There seems to be no end to the possible configurations of both inner and outer servo loop configurations, each with its own advantages and disadvantages affecting system response, stiffness, failure mode, etc.

H) The LVDT was by far the most used feedback sensor. The prevailing opinion was that this was the most reliable and versatile component of the system. Digital encoders are available but are not being used in primary flight control applications.

I) Foreign markets (Japan and Europe)provide the majority of actuation systems capable of accepting direct digital inputs, which are primarily used in the numerical control field.

J) A linear power actuator controlled by a spool and sleeve valve was the most dominant configuration both in use and in development.

K) Electronic logic and the practice of mechanizing the logic functions in the electronics rather than in the actuators was preferred by most participants. Some development work is still being accomplished utilizing hydraulic logic.

L) A substantial amount of effort is being directed toward the development of Integrated Actuator Packages (IAP). These include electro-mechanical, electric motor driven hydraulic servo pumps in simplex, duplex and triplex configurations.

SECTION 3

COMPARATIVE ANALYSIS

3.1 INTRODUCTION

A comparative analysis of the fundamentally different approaches to digital flight control actuation systems was made to select the most appropriate approach for development in the NASA digital fly-by-wire experimental program. The relation of this effort to the total program effort is depicted in Figure 3.1-1. The method used in this analysis consisted of the following steps: (1) performance requirements were projected for next generation aircraft, (2) a general review and evaluation of components and techniques found appropriate for digital flight control actuation systems was made, (3) four approaches incorporating the most appropriate and fundamentally different principals were selected for more detailed consideration, and (4) a trade-off comparison was made between the selected approaches. The four approaches compared in the trade-off analysis were: (1) PM Torque Motor Direct Drive, (2) Stepper Motor Direct Drive, (3) Secondary Actuator, and (4) Electro-Mechanical.

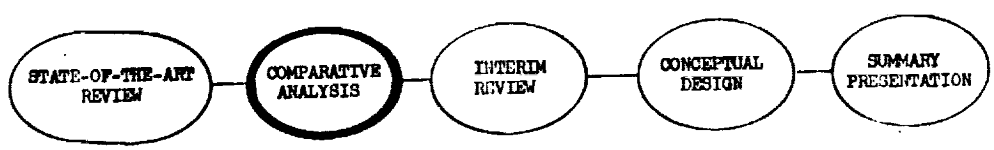


FIGURE 3.1-1 DIGITAL FLIGHT CONTROL ACTUATION SYSTEM PROGRAM

3.2 REQUIREMENTS

Actuation system requirements were established as part of the study to form a baseline upon which to compare the various FBW approaches; these requirements are summarized in Table 3.2-1. Two sets of requirements were established, one for high performance type of aircraft and the other for modern commercial transport type aircraft. Requirements were set sufficiently high to accommodate the needs of future aircraft employing advanced control laws yet not overly stringent so as to introduce an improper bias into the comparison. This paragraph discusses the basis for these requirements.

Flight control systems need to be as simple, direct and foolproof as possible with regard to design, operation, inspection, and maintenance. This requirement, as defined by Military Specification, MIL-F-18372, Flying Qualities of Piloted Airplanes, continues to be an excellent design philosophy for primary controls and needs to be stressed in evolving fly-by-wire systems.

TABLE 3.2-1 PERFORMANCE REQUIREMENTS

	HIGH PERFORMANCE				TRANSPORT			
	HORIZONTAL	RUDDER	AILERON	SPOILER	ELEVATOR	RUDDER	AILERON	SPOILER
FORCE kN (lbs.)	222. (50,000)	89. (20,000)	133.5 (30,000)	66.7 (15,000)	177.9 (40,000)	66.7 (15,000)	35.6 (8,000)	89. (20,000)
STROKE mm (in.)	203.2 (8.0)	203.2 (8.0)	101.6 (4.0)	76.2 (3.0)	203.2 (8.00)	203.2 (8.00)	152.4 (6.00)	152.4 (6.00)
RATE mm/s (in/sec)	254.0 (10.0)	254.0 (10.0)	254.0 (10.0)	254.0 (10.0)	203.2 (8.0)	152.4 (6.0)	203.2 (8.0)	203.2 (8.0)
POWER kW (Hp.)	21.6 (29.0)	8.7 (11.7)	13.1 (17.5)	6.5 (8.7)	14.0 (18.7)	3.9 (5.2)	2.8 (3.7)	6.9 (9.3)
BANDWIDTH (rad/s)	13.5	7.7	7.7	7.7	2.2	2.2	2.2	2.2
RESOLUTION	.1%	.1%	.1%	.1%	.1%	.1%	.1%	.1%
BACKLASH	0.3%	0.3%	0.3%	0.3%	0.3%	0.3%	0.3%	0.3%
FAILURE RATE (FPFH) $\times 10^6$	.53	.53	.53	.53	.053	.053	.053	.053
QUANTITY/AIRCRAFT	2	1	2	6	4	2	4	8

3.2.1 POWER REQUIREMENTS.

Baseline force, stroke, rate, and power requirements were established by surveying four typical high performance aircraft and three present generation transport aircraft: the F-14, F-4, F-15, RA-5C and the DC-10, L-1011, and 747 aircraft. Data on primary actuators utilized in these aircraft are presented in Tables 3.2-2 and 3.2-3.

TABLE 3.2-2 PRIMARY ACTUATORS-HIGH PREFORMANCE AIRCRAFT

	Horizontal					Ailerons					Spoiler					Rudder				
	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)
F-14	2	540.9 (121,600)$_{6}$	222. (8.75)	161. (6.35)	33.5 (45.0)						2	22.7 (5,100)$_{1}$	52. (2.06)	213. (8.4)	1.9 (2.5)	2	97.4 (21,900)	108. (4.25)	179. (7.05)	6.7 (9.0)
											2	29.6 (6,660)$_{1}$	40. (1.56)	160. (6.3)	1.8 (2.4)					
											2	21.8 (4,890)$_{1}$	40. (1.56)	164. (6.46)	1.4 (1.8)					
F-4	2	161.2 (36,240)$_{6}$	267. (10.50)	217. (8.55)	13.5 (18.1)	2	213.1 (47,900)$_{8}$	56. (2.2)	310. (12.2)	25.5 (34.1)	2	33.6 (7,550)$_{6}$	100. (3.94)	446. (17.5)	5.7 (7.7)	1	7.1 (1,590)	59. (2.32)	164. (6.45)	.4 (0.6)
											2	50.7 (11,450)$_{6}$	72. (2.83)	262. (10.3)	5.1 (6.9)					
RA-5C	2	185.3 (41,670)$_{6}$	203. (8.00)	147. (5.8)	10.5 (14.1)						2	55.2 (12,400)$_{7}$	101. (3.96)	246. (9.7)	5.2 (7.0)	1	87.6 (19,700)$_{5}$	206. (8.09)	192. (7.56)	6.5 (8.7)
											2	64.1 (14,400)$_{7}$	81. (3.19)	162. (6.37)	4.0 (5.3)					
											2	64.1 (14,400)$_{7}$	81. (3.19)	162. (6.37)	4.0 (5.3)					
F-15	2	187.7 (42,200)$_{6}$	201. (7.9)	203. (8.0)	14.7 (19.7)	2	140.8 (31,650)$_{5}$	35. (1.39)	638. (25.1)	34.5 (46.3)						Hydraulic Rotary (Vane)				
F-8	2	182.5 (40,980)$_{5}$	144. (5.65)	108. (4.25)	6.4 (8.6)	2	63.7 (14,330)$_{5}$	216. (8.52)	229. (9.0)	5.6 (7.5)	2	27.7 (6,220)$_{3}$	81. (3.17)			1	47.2 (10,620)$_{4}$	81. (3.2)	238. (9.35)	4.3 (5.8)

NOTES:
1. Single Balanced
2. Single Unbalanced - Min Force
3. Single Unbalanced - Max Force
4. Tandem Balanced
5. Tandem Unbalanced - Min Force
6. Tandem Unbalanced - Max Force
7. Parallel Unbalanced - Max Force
8. Quad Parallel Unbalanced - Max Force

TABLE 3.2-3 PRIMARY ACTUATORS-TRANSPORT AIRCRAFT

		Elevators					Ailerons					Spoiler					Rudder				
		Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)	Q	Force kN (lbs)	Stroke mm (in.)	Rate mm/sec (in/sec)	Power kW (hp)
DC-10	Inbd	2	164.6 (37,000)	132 (5.2)	185. (7.3)	11.7 (15.7)															
	Outbd	2	109.2 (24,500)	122 (4.8)	175. (6.9)	7.4 (9.9)															
L-1011		4	81.7 (18.360)	711. (28.9)	356. (14.0)	11.2 (15.0)	6	32.9 (7,400)$_{1}$	138 (5.43)	277. (10.9)	3.5 (4.7)						2	65.4 (14,700)$_{5}$	191 (7.52)	110 (4.32)	2.7 (3.7)
							4	54.1 (14,400)$_{6}$	78. (3.07)	122. (4.81)	3.0 (4.04)										
747	Inbd	2	216.3 (48,620)$_{4}$	206. (8.1)	213. (8.4)	17.7 (23.8)	2	162.4 (36,500)$_{4}$	134. (5.27)	142. (5.6)	8.9 (11.9)	2	93.8 (21,100)$_{3}$	147. (5.79)	234. (9.2)	8.4 (11.3)	1	117.9 (26,500)$_{4}$	137 (5.4)		
	Outbd	2	81.7 (18,350)$_{4}$	78 (3.06)	170. (6 7)	5.4 (7.2)	2	51.2 (11 500)$_{4}$	87. (3.42)	122. (4.8)	2.4 (3.2)	8	65.4 (14,700)$_{3}$	116. (4.57)	184. (7.25)	4.6 (6.2)	1	117.9 (26,500)$_{4}$	137. (5.4)		
C-5A	Inbd	4	110.4 (24,830)$_{3}$	172 (6.77)	122. (4.8)	5.2 (6.95)	4	105.6 (23,750)$_{2}$	159. (6.27)	113. (4.46)	4.6 (6.2)	18	195.5 (43,940)$_{6}$	104. (4.10)	363. (14.3)	27.3 (36.6)	2	56. (12,600)$_{2}$	211. (8.[illegible])	147. (5.7[illegible])	3.1 (4.25)
	Outbd	6	29.3 (6,590)$_{3}$	134. (5.29)	93. (3.67)	1.1 (1.41)											2	41.2 (9,270)$_{2}$	211. (8.36)	14[illegible]. (5.72)	2.3 (3.1)

NOTES:
1. Single Balanced
2. Single Unbalance - Min Force
3. Single Unbalanced - Max Force
4. Tandem Balanced
5. Tandem Unbalanced - Min Force
6. Tandem Unbalanced - Max Force

The size of primary actuators is dictated by the larger of two requirements, maximum hinge moment or stiffness. In sizing it is assumed that all hydraulic systems are operating. The F-14 horizontal was sized by stiffness requirements and is considerably larger than needed for hinge moments. The values selected in Table 3.2-1 assumes hinge moment is the dictating criteria.

3.2.2 PERFORMANCE REQUIREMENTS. Actuation system performance is dictated by its basic function, that is, control of the aircraft. The basic measurement of aircraft controllability is handling qualities or flying qualities. Flying qualities of aircraft have been the subject of extensive study and experimentation. In particular, Cornell Aeronautical Laboratories' Flight Research Department has made notable contributions toward understanding the problems associated with aircraft flying qualities. In 1967 a thorough review of the subject was performed culminating in revision of the "Flying Qualities" specification, MIL-F-8785B, and publication of Reference (169).*

Response requirements specified in MIL-F-8785B are:

> "Dynamic Characteristics
>
> The response of the control surfaces in flight shall not lag the cockpit control force inputs by more than the angles shown in Table XIII, for frequencies equal to or less than the frequencies shown in Table XIII.

TABLE XIII. ALLOWABLE CONTROL SURFACE LAGS

Level	Allowable Lag Rad: Category A & B Flight Phases	Allowable Lag Rad: Category B Flight Phases
1 & 2	.52 (30°)	.79 (45°)
3	1.05 (60°)	

Control	Upper Frequency Rad/Sec.
Elevator	$\omega_{n_{sp}}$
Rudder & Aileron	ω_{n_d} or $1/T_r$ (whichever is larger)

> The lags referred to are the phase angles obtained from steady-state frequency responses, for reasonably large-amplitude force inputs. The lags for very small control-force amplitudes shall be small enough that they do not interfere with the pilot's ability to perform any precision tasks required in normal operation."

The baseline bandwidth requirements were established by these requirements and by surveying present and projected aircraft to establish the highest aircraft frequencies. The RA-5C, F-4, hypothetical 1980 aircraft used in the Navy DIGIFLIC studies, and the F-8 were used for high performance aircraft requirements and the DC-10 and 747 for transport vehicle requirements. Table 3.2-4 lists the highest frequencies found throughout the respective flight envelopes for the above aircraft.

TABLE 3.2-4 AIRCRAFT CHARACTERISTIC FREQUENCIES

AIRCRAFT	$\omega_{n_{sp}}$	ω_{n_d}	$1/\tau_r$
RA-5C	7.8	3.5	3.9
Digiflic	13.5	7.1	4.3
F-8	6.9	5.3	7.7
F-4	5.8	5.8	3.0
747	2.0	2.0	1.4
DC-10	1.7	1.5	1.4

3.2.3 RELIABILITY. Reliability requirements are particularly important in FBW systems due to the obvious consequence of a failure. The level of reliability specified, in a large part, dictates system complexity by necessitating redundancy, built-in test equipment, and monitors. Considerable effort has been expended by various investigators in establishing realistic reliability requirements. The rationale used in these investigations was generally that present primary flight control reliability is acceptable, therefore, a FBW system should be at least equivalent; an improvement would be desirable. Consequently, the investigation became one of surveying and analyzing existing systems to determine actual achieved reliability. One such study,*(043), conducted on Naval fighter aircraft found the loss per flight hour rate to be 11.6 x 10^{-6} LPFH. The lost rate is further broken down in Table 3.2-5.

* Three digit numbers throughout this report refer to the bibliography in Appendix B.

TABLE 3.2-5 AIRCRAFT LOSS RATE (PRIMARY CONTROLS)

CAUSE	LOSS RATE (LPFH)	
Primary Controls	5.55×10^{-6}	
•Actuators		3.2×10^{-6}
•Control Linkages, etc.		2.3×10^{-6}
Secondary Controls	1.84×10^{-6}	
Hydraulic Supply	3.47×10^{-6}	
Other	$.69 \times 10^{-6}$	
TOTAL	11.6×10^{-6}	

Reference (043) estimates that with current FBW technology, (eliminating most of the control linkage and incorporating rip-stop design) the loss rate could be improved as shown in Table 3.2-6 and suggest a goal of 2×10^{-6} LPFH for high performance military aircraft. Past experience in reliability prediction indicates achieved reliability often differs by a factor of 2 from the predicted value. The established goal is sufficient to accommodate this magnitude error and still provide a system equaling existing equipment.

TABLE 3.2-6 AIRCRAFT LOSS RATE GOAL (PRIMARY CONTROLS)

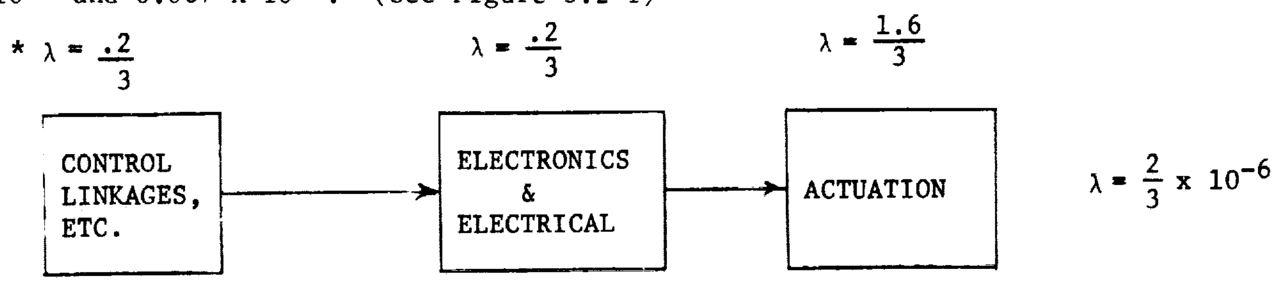
ELEMENT	EXPERIENCED	ESTIMATE	GOAL
FBW Electronic	---	0.1×10^{-6}	0.2×10^{-6}
Power Actuator	3.2×10^{-6}	2.3×10^{-6}	1.6×10^{-6}
Controls, Linkages, Etc.	2.3×10^{-6}	0.2×10^{-6}	0.2×10^{-6}
TOTAL	5.5×10^{-6}	2.6×10^{-6}	2.0×10^{-6}

Similar studies (178, 179 & 180), support these figures and show commercial transport aircraft loss rate to be about one order of magnitude less, thus a goal of 0.2×10^{-6} LPFH. The predicted failure rate for the 680J Survivable Flight Control System was 1.0667×10^{-6} failures per flight hour (177).

The loss rate goals specified are for the entire aircraft primary flight controls which includes the lateral, directional, and longitudinal control systems.

The lateral and directional systems are much more tolerant to failures than the longitudinal systems. Recovery of the aircraft is generally possible with both breaks and hardover failures in the lateral and directional actuation systems except perhaps in a small portion of the envelope or during a critical maneuver. For the sake of expediency it is assumed that the loss rate is equally distributed among the three systems, that the loss of any one system results in loss of the aircraft 50% of the time, and that two actuators per system are utilized.

The failure rate allowable for an actuator is, therefore, 0.53×10^{-6} FPFH and for an actuation system is 0.67×10^{-6} for high performance aircraft; respective values for commercial transports would be 0.053×10^{-6} and 0.067×10^{-6}. (See Figure 3.2-1)



*Failure rate in Failures/10^6 Hrs.

FIGURE 3.2-1 RELIABILITY ALLOCATION

The foregoing discussion is not intended to be an analysis of the reliability problem, but rather a simplistic view to provide some insight and basis for establishment of the actuator reliability requirement. Two obvious discrepancies in the control (FBW) discussion are: first, the primary flight system is now totally dependent upon electrical power therefore the generators, etc., must be taken into account; and second, some aircraft require inertial sensors for stabilization especially for CCV type applications. It is assumed these unreliabilities can be lumped into the electronic and electrical blocks. This may not be the most reasonable assumption, however, it should be remembered that these requirements are established as a baseline for comparative purposes of this report and are not necessarily the final reliability criteria. This loss rate criteria for reliability does not take into account the reliability necessary to achieve a reasonable level of maintainability. Some numbers and goals for maintainability have been established (114). For this study, the relative complexity necessary to achieve the loss rate reliability was used as the measure of maintainability.

3.2.4 ACTUATION SYSTEM STIFFNESS. Actuator stiffness is significant as it relates to flutter. The actuator in essence is one spring in combination with other structural elements, all of which produce resonance. Flutter is normally prevented by assuring that structural resonant frequencies are much higher than the flutter frequency, thus stiff actuators are desired. In some instances stiffness becomes the actuator sizing criteria. A survey, (045), was conducted to determine actuator stiffness requirements for future high performance aircraft and, although a complex subject and dependent upon each specific design, projected requirements were suggested ranging from 6 to 30 x 10^6 lb-in/rad for the horizontal tail and 1 to 4 x 10^6 lb-in/rad for the rudder. The data are reproduced in Table 3.2-7.

TABLE 3.2-7 STIFFNESS SURVEY RESULTS

Airframe Company	Model	Spring Rate MN•m/Rad (Mega Lb-In/Rad) Horiz. Stab.	Rudder	Aircraft Gross T.O. Weight (Kilo Lbs)	Horizontal Stabilizer Hinge Moment (Kilo Lbs-In)
Convair	F-111	3.4 (30.0)	.5 (4.0)	311 (70.0	70.6 (625)
Aerospace	Small Fighter	.71 (6.3)	.17 (1.5)	89 (20.0)	88 (73)
	Projected	1.7 (15.0)	.23 (2.0)	200 (45.0)	--
Lockheed Calif.	F-104	.8 (7.0)	--	89 (20.0)	20.3 (130)
McDonnell/ Douglas	Projected	2.3 (20.0)	.11 (1.0)	240 (54.0)	31.1 (275)
Northrop	Projected	1.1 (10.0)			--
		2.3 (20.0)	Low	111 (25.0)	--
Grumman	F-14	3.4 (30.0)	.14 (1.2)	236 (53.0)	87.7 (776)
					97.6 (864)
Boeing	SST	35.6 (315)		3336 (750.0)	791 (7000)
		16.3 (144)			564.9 (5000)
Rockwell	XFV-12A	1.4 (12 Canard)	.18 (1.6)	89 (20.0)	11.3 (100)
					5.6 (50)

NOTES:
1. Includes all components between actuator/structure interface and control surface.
2. Projected refers to Fighter/Attack Aircraft of the 1975-1980 era.
3. Approximately 50-100 times surface mass moment of interia.
4. Based on a 762 mm (30 in.) horn radius and 5.5 Hz.
5. Based on a 762 mm (30 in.) horn radius and 2.5 Hz (Flutter Frequency).
6. Normal H.M. with 3 of 4 actuators operable.
7. For actuator sizing with 2 of 4 actuators operable and a restricted operating environment.

3.2.5 RESOLUTION AND BACKLASH. Primary control resolution and backlash ultimately establish control precision. Minimum requirements are dependent, among other things, upon aircraft dynamics and control surface deflection sensitivities. There is no generally adequate numerical value to define this requirement. Military Specification, MIL-F-8785, Flying Qualities of Piloted Airplanes, states that freeplay shall not result in objectional flight characteristics particularly for small amplitude control inputs. Military Specification, MIL-F-9490, General Specification for Design, Installation and Test of Piloted Aircraft Flight Control Systems, states residual oscillations shall not produce accelerations greater than the following:

• Normal accelerations in the cockpit	0.02g
• Lateral acceleration in the cockpit	0.01g
• Pitch attitude	1.745 mrad (.1 deg)
• Yaw attitude	2.618 mrad (.15 deg)
• Roll attitude	1.745 mrad (.1 deg)

The requirements of 0.1% resolution and 0.3% backlash were therefore selected from experience. These requirements are not overly stringent yet usually assure satisfactory results.

3.3 COMPARATIVE ANALYSIS DISCUSSION

3.3.1 POWER MEDIA. The fundamental requirement of an aircraft primary power actuation system revolves about the necessary power level; components must function at the required power level and be fully compatible with aircraft power sources and form factors. No matter how reliable a device may be, it has little use if it can't operate the surface. Survey of existing aircraft primary flight control actuation systems disclosed the control power levels shown in Table 3.2-1. High performance aircraft require about 89.5 kW and commercial transports about 134 kW total control power. Actuators range from 6 to 22 kW in high performance aircraft and from 3 to 15 kW in transports. There are four basic system approaches capable of these power levels: mechanical, electrical, hydraulic, and pneumatic. Hydraulic actuation is generally chosen for powered primary aircraft controls for reasons of performance, life, reliability and weight. Mechanical power actuation (i.e., shaft power taken directly from the engine turbine and mechanically transmitted to the surface) has never proved feasible due to controllability problems; electrical power actuation is quite controllable and competitive weight-wise, but can not provide the high performance of hydraulics. Pneumatic systems offer only minor advantages in leakage and replenishment, but have major disadvantages of weight and heat transfer problems due to gas compressibility.

In comparing electrical and hydraulic power distribution system failure modes it is apparent the electrical system has the advantage in controllability. Failure of a pump or alternator in either case results in loss of that entire distribution system unless a second source of power is introduced. Cross-coupling between power systems is easier to accomplish in electrical systems. Cross-coupling hydraulic systems requires a power transfer unit (motor-pump), cross-coupling electrical system requires only bus switching. Electrical faults are easier to detect and isolate. A line failure in hydraulic systems results in complete loss of hydraulic fluid, a permanent loss. "Hydraulic fuses" could be used to detect large leaks enabling fault isolation, however, small leaks are difficult or impossible to detect adequately. Shorts in electrical systems can be protected against, simply and automatically, by circuit breakers appropriately distributed throughout the system to limit loss to individual elements or sub-systems. Weight penalties for redundant electrical transmission systems are lower than for hydraulic distribution systems.

3.3.1.1 Hydraulic Power Actuation. Hydraulic power devices fall within two categories; linear actuators and rotary motors. In the linear category, there are numerous types (i.e., simple balanced and unbalanced, dual tandems, dual parallel, etc.), all of which operate upon the same principle and produce linear outputs. They develop power by producing high force at relatively low rates. In the rotary category, there are also numerous types (i.e., gear, vane, in-line, bent axis, dynavector, etc.), all of which produce rotary output motion. These develop power by producing relatively low torque at high speed. The dynavector is in a sense an exception in that there are no high rotary speed elements, however, the ring gear orbits at high speeds. Reviewing requirements, it can be seen that aerodynamic surface loads are high hinge moment, low rate loads. Inherent characteristics of the linear actuator provide a better match to requirements and can therefore be expected to result in a more efficient design. A comparison is made in the following paragraphs between linear actuators and conventional hydraulic motor driven rotary actuators and then between conventional hydraulic rotary actuators and an orbital motor rotary actuator approach. The comparison is summarized in Table 3.3-1. It is concluded that linear actuators are preferable to rotary actuators except for perhaps specifalized cases where their advantages in travel or form factor outweight their disadvantages.

TABLE 3.3-1 COMPARISON OF LINEAR VS. ROTARY HYDRAULIC ACTUATORS

	LINEAR	ROTARY
Performance	High	Medium
Reliability	High	Low
Weight	Low	High
Stiffness	High	Higher
Travel	Limited	Unlimited
Volume	Higher	Lower
Cost	Low	High

3.3.1.1.1 Rotary vs. Linear Hydraulic Actuators. The requirements of minimum weight and volume, high efficiency, low leakage and high torque holding capability limits consideration of conventional hydraulic motors to the piston types - - bent axis and in-line. There are minor advantages and disadvantages between the two designs, however, these differences are not significant in this comparison. As stated previously, the aerodynamic load is essentially high hinge moment (torque) at relatively low rate. To use high speed motors, a gear reduction box is required, and for dual inputs or redundant drives, a differential is required. To use the linear actuator, a crank is necessary to convert linear to rotary motion. Typical systems constructed with these components are depicted in Figure 3.3-1. Sizing of gears, gear boxes, differentials, etc., is primarily a function of torque. High torque and the corresponding high force levels require sufficient material to maintain stress levels within design limits and fatigue life considerations. The size is only loosely correlated to the power level; small gears at low torque and high speed can transmit high power. To minimize weight in the rotary system, the differential is placed in the high speed low torque region next to the motors. Weight of the rotary actuator is predominately due to the gear box; consequently a torque to weight ratio in the 50-100 N·m/N (2 to 4000 lb-in/lb) range. Component weight data is presented in Appendix F.

Performance capability of the linear system is considerably higher than the rotary system for a number of reasons. Friction in the rotary actuator is high and in addition, there is differential and gear box friction. Friction is composed primarily of stiction and coulomb. Coulomb friction is not particularily troublesome in the inner loop, however, stiction has a very destabilizing effect which degrades performance. Both types of friction introduce phase shift in closed loop response. Coulomb friction in large linear actuators is typically in the 0.5 to 1.5% and stiction in the 0.2 to 1.0% range. Stiction in hydraulic motors, in the 15 to 25% range, limits smooth operation at low speeds. Standard in-line motors operate smoothly down to about 31.4 rad/s (300 rpm). Below 31.4 rad/s (300 rpm) rotation is erratic, often referred to as chugging. Servo motor designs can be made to operate smoothly down to about .105 rad/s (1 rpm). Hydraulic motors perform well in constant high speed operation. In cyclic operation where load reversals occur rapidly extremely high torques and internal pressures develop which must be limited by relief valves to prevent motor failure; this limits high frequency performance.

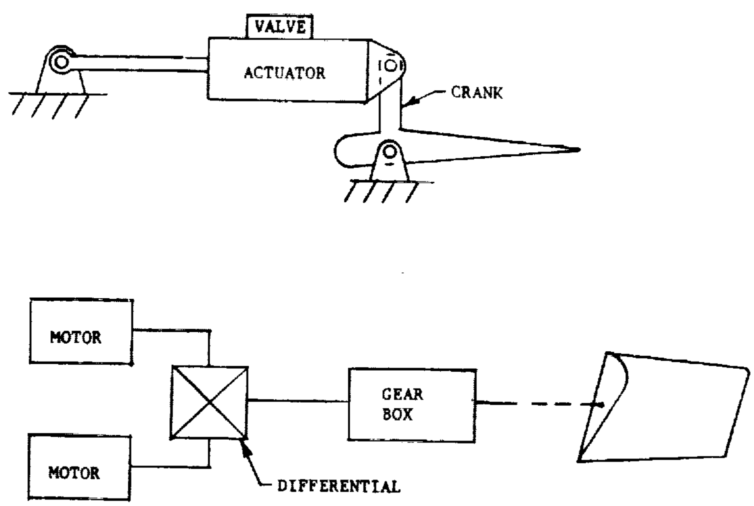


FIGURE 3.3-1 LINEAR AND ROTARY ACTUATION

Reliability of the linear actuation design is considerably higher than the rotary design. Figure 3.3-2 compares reliability of the two designs; the linear system is 3 times more reliable than the rotary. Component reliability data and sources are presented in Appendix E. Relative reliability of the two designs is simply demonstrated by parts count; the rotary system has approximately 3 times as many parts as the linear actuation design.

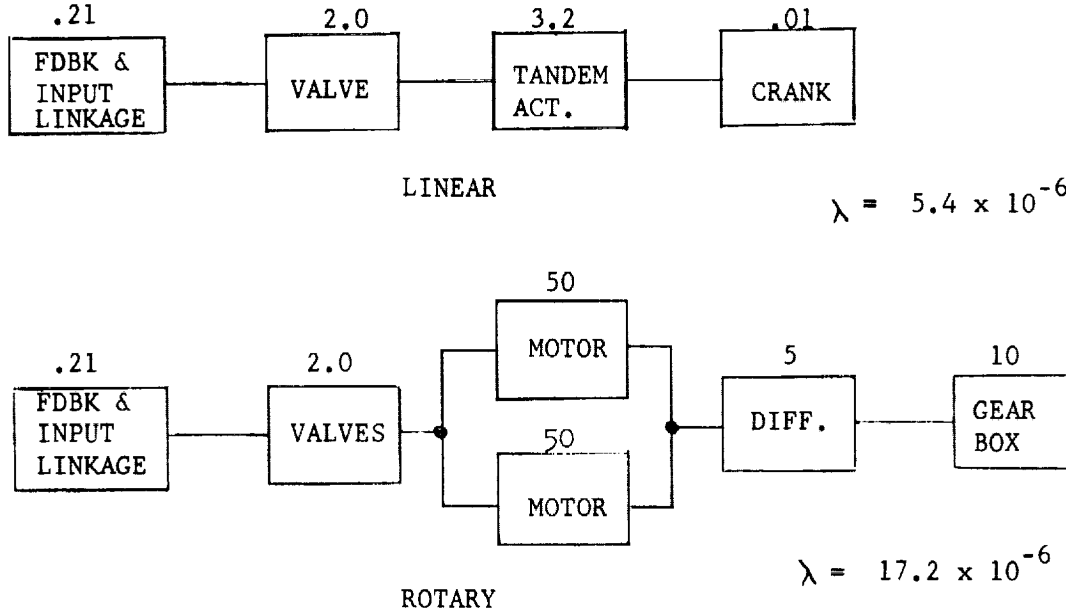


FIGURE 3.3-2 LINEAR & ROTARY ACTUATION RELIABILITY DIAGRAMS

Service life of systems incorporating high speed, high power density components, and high parts count is in general less than systems characterized by low speed and few parts.

The weight of linear actuation is a function of the actuator work figure, force times stroke. As required surface travel increases, kinematics of the simple linear arrangement of Figure 3.3-1 deteriorates, and beyond 1.05 rad (60°) deteriorates rapidly. The moment arm decreases with travel, requiring larger actuators and larger space envelopes due to both the actuator and swept volume. The rotary actuation approach is not limited in stroke, can be packaged in minimum volume and requires no swept volumes. For the surface travels required in conventional aircraft 0.5 to 1.0 rad (30° to 60°) kinematics of the linear system is quite efficient and results in considerably lighter design. The initial cost of linear actuation is considerably less due to the number of precision parts required. The cost of ownership is also less due to parts cost, life, and reliability. The linear actuation system is thus cheaper to purchase and cheaper to own.

3.3.1.1.2 Orbital Motor vs. Hydraulic Motor. The orbital motor (144, 145, 146) actuator is an innovative design integrating a captive vane orbital motor and epicycle transmission which provides high torque at low speed. This combination is comparable to a hydraulic motor and gear box, but offers higher performance, fewer parts, lower friction losses, and higher torque-to-inertia ratio. The orbital motor actuator is capable of being decoupled from the load shaft without high torque clutch elements where dual inputs or drive are required. This eliminates the need for differentials used with hydraulic motors in the same application resulting in fewer parts. Achievement of published (043) weight goals of 50 N•m/N (2000 lb-in/lb) would make it competitive with conventional rotary actuator designs.

The orbital motor actuator is in the early stages of development and, as such, its reliability and life are unproven. Its ultimate application in flight controls remains to be seen.

3.3.1.1.3 High Pressure Hydraulics. Very high pressure hydraulics have been under continuous investigation since 1966. Studies to date indicate the most practical operating pressure is around 55.16 MN/m^2 (8000 psi). An application study (162) performed on the F-14 aircraft showed a 30% reduction in weight and a 40% reduction in the total hydraulic system volume; the 55.16 MN/m^2(8000 psi) system weighed 6.2 kN (1400 lbs) compared to 8.9 kN (2000 lbs) for the 20.68 MN/m^2 (3000 psi) system. Extensive study and laboratory tests indicate 55.16 MN/m^2 (8000 psi) to be a practical limit with present technology. A program is now under way to demonstrate an 55.16 MN/m^2 (8000 psi) system in flight tests, to be conducted in early fiscal 1975. High pressure hydraulics is recommended for future aircraft as a method for reducing hydraulic system weight and volume.

3.3.1.1.4 Hydraulic Supply Systems. A nominal failure rate for hydraulic supply systems, based upon data accumulated for current military aircraft, is 178 x 10^{-6} FPFH. There is of course considerable variation above and below this figure depending upon the specific design, system complexity, etc. If dual systems were completely independent, then an expected dual failure rate would be .031 x 10^{-6}. An aircraft loss rate due to hydraulic system failure of over two orders of magnitude higher, 3.47 x 10^{-6} LPFH is indicated by the same data. Part of the difference can be explained by definition and categorization of the failure, and the remaining portion by dependence upon a common denominator such as fatigue, maintenance, contamination, etc. A realistic failure rate number when using redundant configurations is 1780 x 10^{-6} FPFH. A realistic dual hydraulic system failure rate is 3.16 x 10^{-6} and 5.65 x 10^{-9} FPFH for a triple hydraulic system. A triple hydraulic system is considered completely adequate. A dual hydraulic system is considered satisfactory for high performance military aircraft.

3.3.1.2 Electrical Power Actuation. Electro-mechanical (EM) actuation is utilized in numerous secondary control applications and now in the A-11 leading edge which is a flying surface. Considerable advances in the state-of-the-art have been made which modify previous evaluations; namely, new materials, solid state controllers, improved manufacturing techniques, and computer-optimized designs. New materials, chiefly insulation and metals, enable fabrication of motors which weigh 60% less and occupy 80% less volume than motors of thirty years ago, and it is expected the motors will continue to shrink in size and weight. Laboratory units have been fabricated that operate at 760°C. Solid state motor control technology permits faster response with higher power and rate capability.

3.3.1.2.1 Space Shuttle EM Study. The actuation system depicted in Figure 3.3-3 was studied in considerable detail for application on the space shuttle and was compared against a hydraulic actuation system approach (159). The actuation system (044) consisted of electronic switching modules, A-C induction motors, gear boxes, no-backs, torque shafts, and rotary actuators. Motor control techniques investigated included fixed voltage, fixed frequency, pulse modulation, variable voltage and frequency pulse control, and bang-bang control with and without no-back devices. The most promising technique found was the variable voltage/variable frequency-bang-bang control with no-backs. In the variable power/frequency system, frequency and voltage are both varied with motor speed, resulting in reduced motor losses since the motor operates at lower slip. Motor heating is also reduced during stopping and reversing.

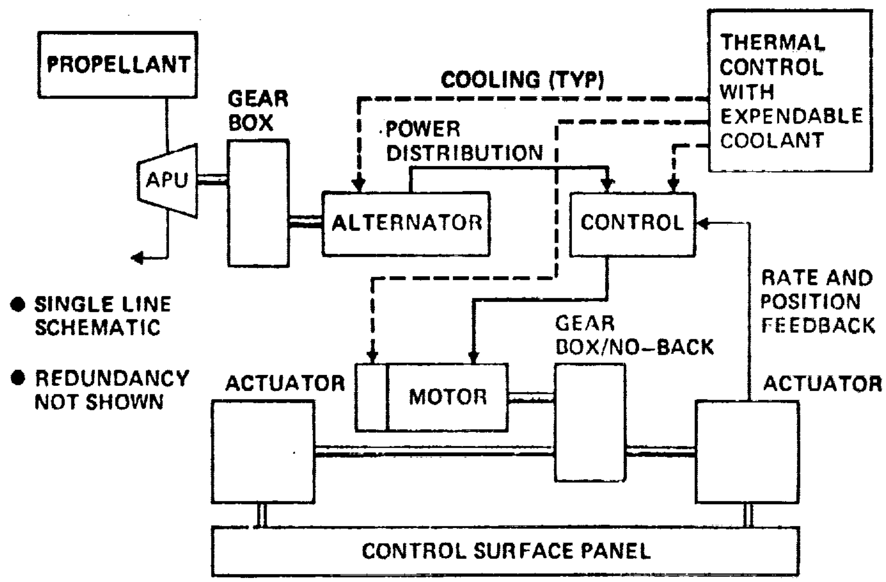


FIGURE 3.3-3 EM SYSTEM ARRANGEMENT

With the bang-bang control system, the motors are either fully on or fully off. An error signal switches the motors on and as surface approaches the commanded position the motor is switched off. The motors in this system are pulsed less per cycle than with the pulse modulated control system. Motor losses and power required are less than the pulse modulated system.

No-backs are used in the electro-mechanical control system to ease the thermal load on the motor and reduce power consumption. For a holding load, the motor is switched off and requires no power. For an aiding load, the motor operates at practically synchronous speed; from a thermal standpoint, this is the most favorable mode of operation.

The Space Shuttle study shows that the EM system is feasible and can be developed to compare favorably in reliability and response with the hydro-mechanical system approach. The EM system is sensitive to duty cycle; increasing the frequency of low amplitude control surface movements results in increased usage and increased power consumption and heating. The EM system requires development of motors, generators, and variable frequency thyristor switching modules. It has also been suggested that EM has advantages in reduced contamination potential, fire hazards, and offers simplified maintenance and repair.

The mission requirements of the Space Shuttle differ significantly from aircraft, especially mission time, environment, and weight penalties. The resulting conclusions are therefore not completely applicable to aircraft, but do strongly indicate that a detailed study of EM application in primary flight controls should be conducted.

Rotary or linear electro-mechanical actuation systems can be mechanized. A typical rotary approach is shown in Figure 3.3-4. The electric motor, like the hydraulic motor, is a high speed, low torque device; high ratio gearing is required to match the load requirements. A differential is required to incorporate redundant drive. A no-back device must be used to achieve zero input power load holding, a feature inherent in piston type hydraulic motors. Without a no-back device, the motor is required to hold steady-state loads (stall condition) which is undesirable from both power consumption and heat dissipation considerations. The EM rotary actuation approach is mechanically similar to the hydraulic rotary approach and likewise the weight is high, and dictated primarily by the gear box.

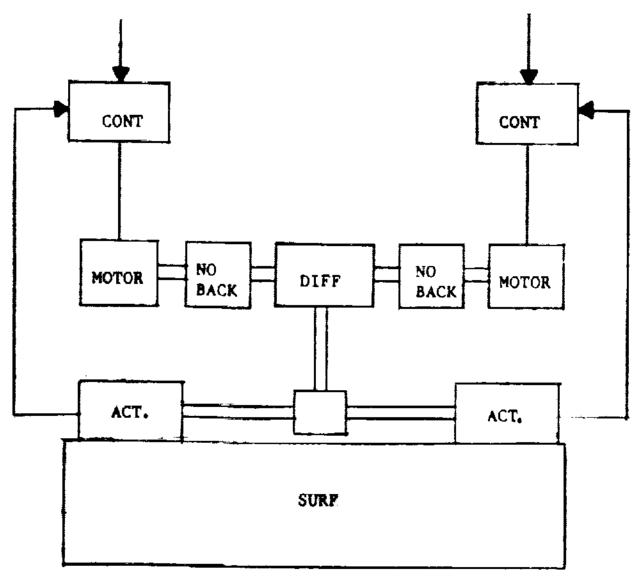


FIGURE 3.3-4 REDUNDANT EM ROTARY ACTUATION

A linear electro-mechanical actuation system is shown in Figure 3.3-5. Redundancy is provided by dual motors; one motor drives, via a no-back and gear box, the screw and the other motor, via the no-back, gear box, and splined shaft, drives the nut. The system would operate in an active/standby redundancy mode. This approach has considerable merit, it is competitive weight-wise with linear hydraulic actuation, and is very stiff. (Reference weight data in Appendix F). With the advances in the state-of-art, this approach appears capable of meeting necessary response requirements, at least for transport vehicles. Considerations of life and heat dissipation require further study.

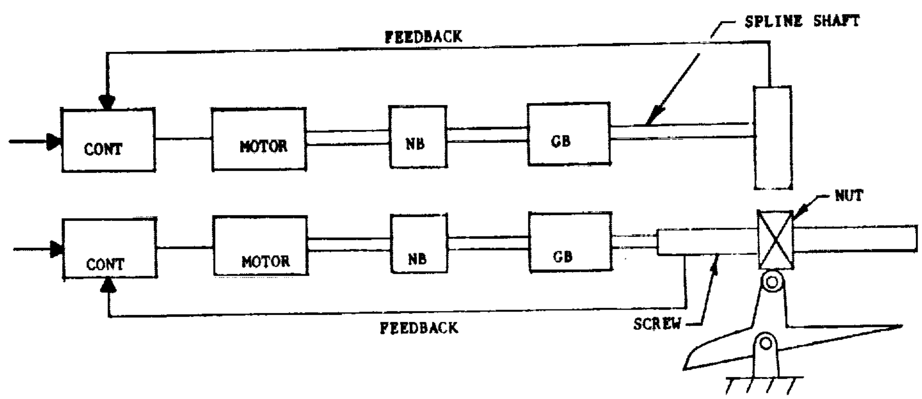


FIGURE 3.3-5 REDUNDANT EM LINEAR ACTUATION

Four typical types of no-back devices are: (1) irreversible components, (2) thrust collars, (3) capstan coils and (4) ramp activated no-backs. Ramp activated designs appear suitable for the EM system, and can be made essentially chatter free when driving with an aiding load. Torque required is approximately ten percent of the load torque.

Another method of accomplishing the no-back function to incorporate an electro-mechanical brake in the motor. The brake is spring loaded to on, requiring current for release. When the motor is not driving, the brake is applied. A delay is ordinarily incorporated to reduce brake cycling during various periods to extend life.

3.3.1.2.2 Brushless DC Motors. Advances in solid state technology have made possible development of electronic circuits which, when combined with a properly designed motor produce an electro-mechanical converter corresponding in operation to a conventional DC motor. This package offers an advantage over the conventional DC motor by eliminating brushes with their inherent limitations. It offers advantages over A C induction motors since it has a speed-torque curve which more closely matches the loads required in moving flight control surface; i.e., higher torque at low speeds.

At present, brushless DC motors are limited to one to 0.7 to 1.5 Kw (1 to 2 hp) maximum which is insufficient for powering most primary flight control applications. For this reason the brushless DC motor is not considered further in this study. High voltage DC power systems are being investigated for aircraft applications. The use of a high voltage DC power supply would increase the capability of the brushless DC motor to as much as 5.2 kW (7 hp); this would be adequate to drive many primary flight control surfaces. It is believed that with further advancement in the state-of-the-art, more powerful brushless DC motors will become available and at some future date could be feasible for powering primary control surfaces.

3.3.1.2.3 <u>Electrical Power Supplies.</u> The loss rate of an electrical power system, based upon military aircraft data, is 128 x 10^{-6} LPFH. If it is assumed there is a common failure cause factor among redundant electrical systems, as in hydraulics, then a realistic rate to use in redundant schemes is 1280 x 10^{-6} FPFH. Total failure of a triple redundant power system is therefore 2.1 x 10^{-9} FPFH; this is considered adequate.

3.3.1.2.4 <u>Integrated Actuator Packages (IAP).</u> Integrated actuator packages are basic components in power-by-wire applications. The package is powered electrically and consists of a motor driven hydraulic pump which supplies hydraulic power for a surface actuator. Two types have been built: a constant supply system (026) and a demand system (027, 048) or servo pumps. A major problem in IAPS is heat. Power wasted due to inefficiencies results in heat which is difficult to remove. The servo pump approach is the most efficient and therefore the most practical. IAP's offer advantages in power system fault detection, isolation afforded by electrical power, reduced contamination problems, minimization of hydraulic leakage, and reduced total power consumption. Weight savings are possible, especially small power applications when compared to centralized hydraulic systems. Packaging is cumbersome and requires larger mounting space at the control surface. In comparison with the EM, power-by-wire approach popularized by Space Shuttle work, the obvious advantage is in performance since an IAP is not dependent upon accelerating an electric motor. Initial and maintenance cost would probably be higher. Weight is estimated to be comparable to EM systems. Both utilize electric motors for power conversion, one uses a mechanical transmission and actuator, the other a hydraulic transmission and actuator.

One further observation -- contamination potential should be reduced by sealing the packages at manufacture, and requiring no field maintenance. Only one external seal is used and this can be redundant. Packages needing repair should be returned to the manufacturer for re-work under ideal clean room conditions. Two stage servo valve reliability will be high under these conditions. The conventional two stage servo valve is directly compatible with advanced electronic technology, CMOS, PMOS, and NMOS, which appears to be the trend for the future. This compatibility eliminates the need for the power servo amplifier and ancillary equipment such as power supplies. The D/A conversion can easily be accomplished in the valve. Autonetics Division of Rockwell International has been actively studying the digital computer hydraulic actuation interface problem for missile applications. A number of their advanced concepts in this area are directly applicable to aircraft.

3.3.1.3 Mechanical Power Actuation. A mechanical power actuation system receives shaft transmitted power directly from the prime power source. A control element such as an electrical servo clutch controls the power input to a mechanical actuator. There are two types of mechanical actuators; rotary (power hinge) and linear (jack-screw). The two types are depicted in Figure 3.3-6. Mechanical actuators are reliable devices and have been used extensively in secondary applications. Rotary systems are heavy due to the gear box which must carry full hinge moment torques. The linear mechanical actuation system is lighter in weight, and competitive with linear hydraulic actuators. The linear mechanical actuator has a number of desirable features; light weight, simple, reliable and rugged. The gear box, since it is in a low torque area of the system, is fairly light and reliable. Both linear and rotary mechanical actuators are very stiff - - stiffer than hydraulic actuators. The linear type has considerable weight advantage over the rotary but is limited in travel by kinematics. Backlash in rotary actuators is typically 4 minutes of arc and can be reduced by precision gears or preloading. The control device required is some form of an electro-mechanical clutch. Unfortunately clutches do not meet primary flight control reliability or life standards and as such are not suitable for primary control applications. The survey effort did not uncover any new developments that would modify this conclusion. The linear mechanical actuator, however, is a good device and is used extensively in secondary controls where it is driven directly by a controllable electric, hydraulic, or pneumatic motor.

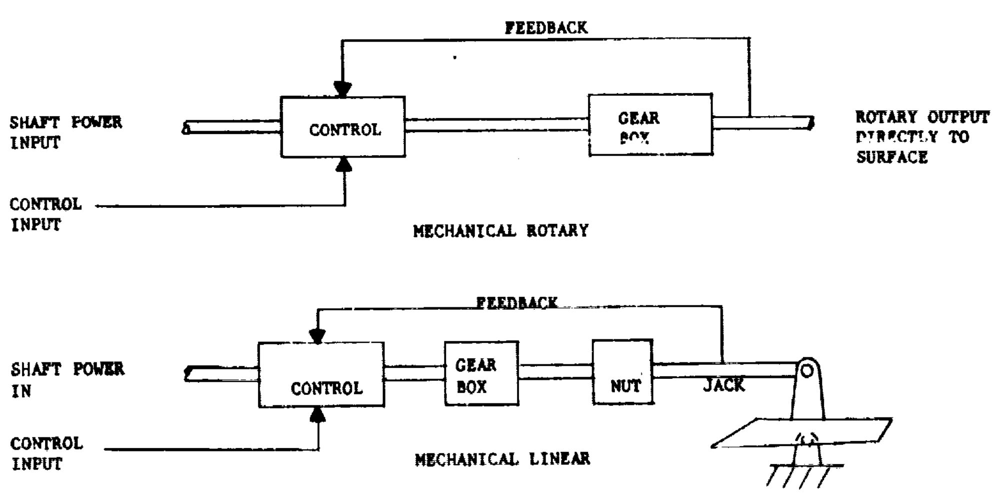


FIGURE 3.3-6 MECHANICAL ACTUATION SYSTEMS

3.3.1.4 Pneumatic Actuation. The survey effort did not disclose any new developments in pneumatics that would modify the conclusions of reference (069); namely, that some development work has been accomplished on pneumatic servoactuators but very little hardware is in existence. The work on penumatic development has been essentially to advance the state-of-the-art of pneumatic servos to the level that hydraulics reached nearly 20 years ago, as such, there is no specific correlation between pneumatic servo development and fly-by-wire development. It was noted that two air motors are employed on the Harrier V/STOL aircraft in the engine nozzle actuation system to furnish power for positioning engine nozzles. This development is unique to the Harrier V/STOL aircraft. Air motors are also used extensively in engine control applications characterized by low power and high temperature. A pneumatic orbital motor unit is being developed for power boost and SAS applications utilizing .34 MN/m^2 (50 psi) air.

3.3.2 POWER CONTROL. Two power media are considered appropriate for primary flight controls, hydraulics and electro-mechanical. This section briefly discusses power control devices and approaches found in the survey.

3.3.2.1 Hydraulic. The survey identified the following representative hydraulic digital and analog devices:

a. Digital Devices

1. Electro-Hydraulic Linear Stepper Actuator (141)
2. Jo Block Actuator (003, 073)
3. Digitizer Valve (049)
4. Stepper Motor Valves (161)
5. Two Stage Stepper Motor Valve (166)
6. Digital (Analog) Servo Valves (049)
7. Saturation Hydraulic Valve (082)
8. Summing Hydraulic Valves (107)

b. Analog Valves

1. One and two stage jet pipes
2. One and two stage flapper nozzle
3. PM torque motor direct drive
4. Linear force motor direct drive

3.3.2.1.1 Digital Devices

a. Electro-Hydraulic Linear Stepper Actuator

The underlying principals of this actuator are depicted and discussed in Reference (141). The control valve arrangement could be a rotary valve with ports indexed to a stepping motor or could be simple solenoid valves actuated sequentially. Valving could be located remotely from the actuator.

Some limitations of the actuator may be derived by applying the concept to a typical design:

1. Total actuator length is over twice the stroke length.

2. Orifice size is small stemming from the resolution requirement (0.3%). 0.3% of a typical stroke of 203.2 mm (8 in.) is .69 mm (.027 in.). Porting arrangements can possibly be designed utilizing two to three times this size, however, this still would be undesirably small. In defense it is noted that clogging of an orifice does not cause hardovers; it would either fail dead or lose resolution depending upon where the clogging occurs.

3. The design is complex to manufacture due to the number of precision orifices which must be machined in the receiver. Considering six transmitter valves, 56 orifices are required to give 0.3% resolution over an 203.2 mm (8 in.) stroke. This would require 7 valves for communicating and direction controls.

4. As in any sequential open loop approach, the actuation element must be relatively insensitive to load, otherwise synchronization would be lost if the rate command exceeded the system capability. A feedback sensor could be added for monitoring and information purposes (only) without direct operation in the control loop.

5. The total force capability of the actuator is a function of the differential area and is unbalanced.

6. Stiffness is poor and is soft in one direction since there is no hydraulic lock.

The actuator is novel and has been patented in France with applications filed in the United Kingdom, Europe and USA. The manufacturer is seeking applications and/or licensees.

The principal advantage of the actuator lies in its ability to move incrementally over small steps with high accuracy without feedback. Because the actuator and its associated controls are fairly simple, it could be considered for applications where resolution is not severe, strokes are not long, and output forces are not large. It is not considered applicable for flight control power actuators.

b. The "Jo Block" Actuator

The "Jo Block" actuator [073, 003] approach is not considered feasible for primary control applications because of: (1) complexity, (2) size and weight, and (3) dynamic response. To obtain adequate resolution, 10 elements are required which in turn require 10 valves and 10 electronic drive circuits. Careful design with perhaps considerable sophistication is necessary to obtain proper dynamic response of the combined elements. For example, assume the actuator is positioned at the most significant bit position and a command is presented to move to one bit less. All ten elements must move; the most significant bit must move to the 0 position and all other must move to their 1 positions. Dynamic movement must be controlled fairly accurately otherwise the smaller bits could get in before the large bit is removed and the actuator temporarily moves in the wrong direction. This type of unit is exceedingly noisy due to clanking of the elements and water hammer transients which produce high vibration levels. Serial stacking of the elements results in an exceedingly long actuator for the stroke provided. The weight is a problem due to the relatively inefficient design. This approach has advantages of parallel inputs, quite accurate positioning, and insensitivity to loading (i.e., cannot loose synchronization). There is a degree of inherent redundancy; valve failure either open or closed does not result in hardovers or loss of control, but rather results in degraded performance.

This approach has a number of features desired in a digital actuator. Unfortunately the physical design is unsuitable for applications in either primary or secondary actuator applications in flight control systems.

c. Digitizer Valve

One approach to obtain a digital actuator is by introducing a fixed quantity of fluid into one actuator chamber and removal of a like quantity from the other. Several approaches have been investigated and some even built and tested [143, 184]. Performance attainable is severely limited by: (1) the cycle time required to inject the fixed quantity of fluid, and (2) the output is not absolutely synchronized to the input due to leakage, thus an additional synchronization or feedback means is required for long term operation.

The digitizer approach is not considered feasible for primary control applications.

d. Stepper Motor Valves

The electrical stepper motor, a digital electrical to mechanical transducer, has been integrated with the spool/sleeve type valve to interface digital system with hydromechanical systems [161].

Stepping motors provide controlled rotational steps or essentially digital position response to an input of sequential pulses. A common use in industry today is in open loop positioning systems that can replace, in many instances, more expensive and complex closed-loop servo systems. Stepping motors can start rapidly, slew at high rates, and stop almost instantaneously without the use of brakes or clutches. These devices can also be used effectively in closed loop servos.

The basic types of stepping motors suitable for primary control applications are: (1) variable reluctance (VR) and (2) permament magnet (PM). Both types are brushless and of relatively simple construction with the rotor and bearings being the only moving parts. Characteristics of the VR motor are:

1. Simpler in construction than PM
2. Low rotor inertia
3. High speed capability
4. Rotate freely when not energized
5. Step sizes are usually .13, .26, or .52 rad (7.5, 15, or 30 degrees)

Characteristics of the PM type are:

1. More power from given frame size
2. More efficient than VR
3. Better damping than VR
4. Has a holding torque when not energized
5. Step sizes typically are .03, .04, .13, or .26 rad (1.8, 2.5, 7.5, or 15 degrees)

Most stepping motors are capable of bi-directional operation. Stepping rates vary typically from 160 steps per second to 10,000 steps per second depending on motor size, step size, drive circuit capability, and load characteristics.

The stepping motor rotates in response to a changing pattern of interactions between the rotor and stator magnetic fields. A drive circuit is required to create the proper sequence and power levels of pulses for application to the stator. Overall performance of the stepping motor system is heavily dependent on the drive circuit, not only in available power delivered to the load, but also in such parameters as efficiency, power dissipation and cost. The drive circuit typically consists of one to three printed circuit cards with solid state components plus a power supply.

The stepping motor should be considered a candidate for controlling the hydraulic power actuation system in a digital fly-by-wire flight control system. Hydraulic pulse motors are being utilized extensively in foreign numerical controlled machinery designs. Advantages touted are system simplicity, accuracy, and maintainability. The heart of the design is the electrical stepper motor coupled to a hydraulic motor in a follow-up mode (mechanical feedback). In essence the electrical stepper motor drives a spool valve via a screw. The valve then controls rotation of the hydraulic motor which follows-up and drives the spool back to null. Units are commercially available in sizes up to 7.46 kW (10 hp) and 167.6 rad/s (1600 rpm).

A significant advantage of the stepping motor is that it is inherently a digital device and as such fits well into a digital system. Disadvantages include complexity of the drive circuit required, necessity for gearing down to obtain required resolution, necessity for rotary to linear motion conversion, and complexity of coupling two or more motors together to achieve redundancy.

Two types of stepper motor driven valves were found in the survey -- single stage and two stage [161, 166]. The single stage type, common in numerical control, is depicted in Figure 3.3-7. The two stage type, a rather unique approach in rotary to linear motion conversion, is shown in Figure 3.3-8. This valve [166] was developed for the Marshall Space Center by HLM, Inc. The main advantage of the two stage approach is the high shear out force capability provided by hydraulic power; the stepper motor is required only to open the valve and as such must only work against friction and silting. The response of this particular design is inadequate for primary control applications, however, with the proper trade-off between performance parameters, it is believed response could be raised.

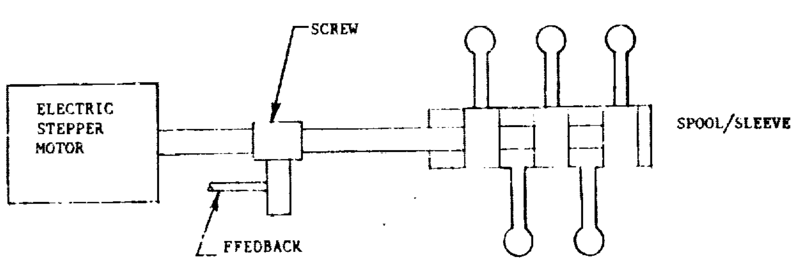


FIGURE 3.3-7 SINGLE STAGE STEPPER MOTOR VALVE

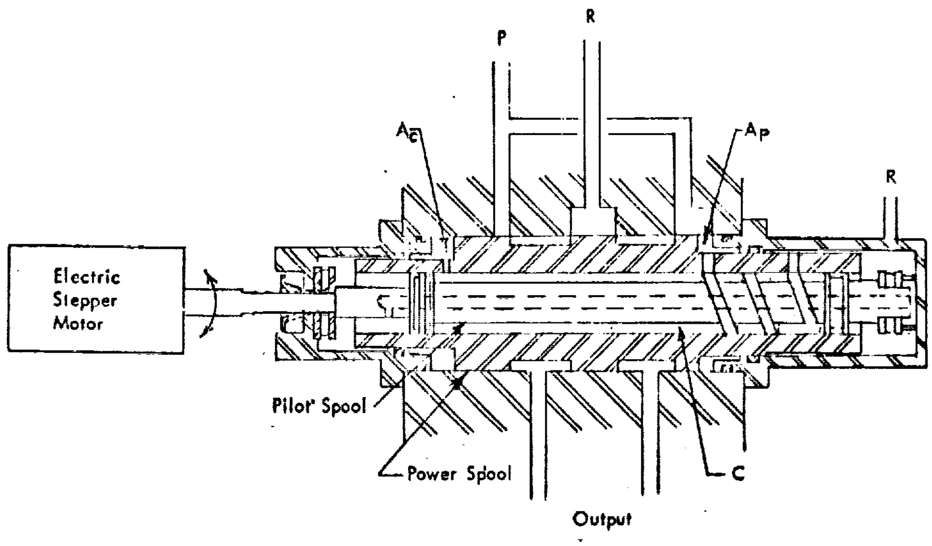


FIGURE 3.3-8 TWO STAGE STEPPER MOTOR VALVE

e. Digital (Analog) Servo Valves

Conventional servo valves (i.e., two stage jet pipe, flapper-nozzle, etc.) can be operated with digital inputs thereby accomplishing in part the D/A conversion. Three basic approaches were investigated, all of which involve some form of summation: (1) input current summation, (2) input torque or flux summation, and (3) pulse width modulation. Input current summation accomplishes D/A conversion by a resistance ladder network using the servo valve as a current summing amplifier. Torque summation is effected in the torque motor where D/A conversion is accomplished using multiple coils, in which each successive coil has either an equal number of turns or twice as many turns as the previous coil (binary progression).

In the equal turns design, each successive coil is driven digitally by twice the current as the previous coil. The pulse width modulation design accomplishes D/A conversion by modulating a carrier with the digital input command and applying the modulated signal to the valve coil -- or in one example a crystal flapper driver [168]. The valve is unable to follow the high frequency carrier and operates in accordance with average value. Although these designs operate in a digital manner they are dependent upon precision references and do not have the non-precision advantage associated with digital circuits. The low current requirements of conventional servo valve simplify the driving circuitry. This permits the use of MOS components to drive the valve directly and eliminates the need for servo amplifiers and power supplies. The main objection to this scheme is the contamination sensitivity and inefficiency of the two stage electro-hydraulic servo valve.

f. Saturation Hydraulic Valve

One approach [082] which circumvents the need for precision references when conventional servo valves are digitally operated utilizes 8 small solenoids in place of the valve torque motor. The solenoid magnets are on a common shaft which works against a spring and positions the flapper in two-stage, nozzle/flapper valve. This arrangement provides a force sum of the solenoid magnets. The magnets operate in the saturation region and therefore precision electrical power supplies are not required. A 7 bit valve requires 8 magnets. This design is complex and cumbersome, and is sensitive to contamination due to the flapper/nozzle stage.

g. Summing Hydraulic Valves

An approach [166, 107] to designing a digital servo valve is to sum the flow output of multiple solenoid valves. Hardware required to produce one 7-bit valve includes one three-way valve and seven on-off type valves. The amount of hardware, although providing some redundancy, is considerable; this aspect eliminates the approach from further consideration.

Simplification can be had by reducing the number of bits. A one bit design (two valves or one two-way valve) could be used in a bang-bang control loop. Poppet valves are reliable, rugged devices, but it is doubtful whether primary control servo loops can be constructed having adequate performance, life, and efficiency. Bang-bang control is used extensively in missile applications where conditions of low cost and short life prevail. One typical application is in pulse-width-modulation where quasi-linear operation is obtained at the expense of life, power, and stiffness. Of course, any analog valve can be used for bang-bang control.

3.3.2.1.2 Analog Servo Valves. There are four basic types of servo valves: flapper nozzle, jet pipe, deflector jet, and direct drive spool/sleeve valves. The first three are in common use and characterized by high response and performance, low input electrical power, low flow capability, high quescent leakage flow, and small delicate parts. They use small PM torque motors for electrical to mechanical transformation. High flow capability is achieved by addition of a second stage spool/sleeve. Direct drive spool/sleeve valves are less commonly used and are characterized by single stage, rugged parts, low leakage flow, lower response, higher electrical input power, and high flow capability. A number of electro-mechanical transducers can be used to drive the spool: PM torque motors, moving coils, plunger magnets, DC torque motors, stepper motors, etc. The PM torque motor is preferred because of its high torque-to-inertia ratio, low electrical input power per unit torque output, simple design, and few parts. The crucial characteristic in single stage spool/sleeve valve designs is development of sufficient force to shear out contaminants. Investigations [114, 183] discloses that force levels comparable to those used in mechanical flight controls are now possible in single stage valves due to the advances in solid state electronics. The first stage hydraulic amplifier is no longer mandatory.

A comparative evaluation was made of four conventional servo valve designs and three uncommon servo valve designs for use in FBW actuation systems. Specific parameters listed under four basic characteristics -- performance, reliability, weight, and cost -- were rated poor (1-3), good (4-7), and preferred (8-10). Of the four basic characteristics cost and weight are considered of secondary importance, since valve weight is generally a small percentage of total actuation system weight and cost is fairly reasonable for all the valves. The main criteria upon which to base the selection is reliability and performance. Results of this comparison are presented in Table 3.2-3.

TABLE 3.3-2 SERVO VALVE COMPARISON

	JET PIPE	FLAPPER NOZZLE	2 STAGE JET PIPE	2 STAGE FLAPPER NOZZLE	2 STAGE STEPPER MOTOR	1 STAGE PM TORQUE MOTOR	1 STAGE LINEAR FORCE MOTOR	STEPPER MOTOR VALVE
PERFORMANCE								
RESPONSE	10	10	8	8	5	9	9	5
HYSTERESIS	8	8	8	8	7	7	7	7
NULL STABILITY	6	6	6	6	8	7	7	8
LINEARITY	9	9	9	9	7	8	7	7
LEAKAGE	3	3	3	3	9	9	9	8
ELEC. POWER	9	9	9	9	4	6	3	4
RELIABILITY								
RUGGED	7	7	6	6	8	10	10	8
CONTAMINATION TOLERANCE	8	7	7	6	10	9	9	9
DESIGN SIMPLICITY	8	7	7	6	9	10	10	8
FAILURE MODES AND EFFORTS	7	6	6	5	9	9	9	8
LIFE	7	7	6	6	8	9	9	8
WEIGHT	10	10	9	9	3	4	3	3
COST								
INITIAL	8	8	7	7	4	4	6	4
MAINTAINABILITY	4	4	3	3	7	8	7	7

The comparative analysis indicates the PM torque motor direct drive design is the first choice followed by one and two stage stepper motor designs. Presently designed two stage jet pipe and flapper nozzle valves are not rugged enough nor tolerant enough to contamination. They also suffer from high power losses due to leakage which is further aggravated at VHP levels. The two stage stepper motor valve compared to the PM torque motor direct drive approach has the advantage of higher shear out capability, but has disadvantages of lower frequency response (limited by the stepper motor, complex drive circuitry, and a non-null electrical failure mode (the stepper motor remains at its last position when de-energized). The non-null failure mode could possibly be circumvented with mechanical feedback or some other technique with attendant increase in complexity. The PM torque motor direct drive valve is of conventional design although requiring a larger motor than now commercially available. A model is under test and development utilizing high strength cobalt samarium magnet material which is particularly well suited for this application [183]. One and two stage stepper motor valves (although fairly straight forward) are not presently commercially available as separate units and require development.

3.3.2.2 <u>AC Motor Control</u>. AC induction motors inherently have low starting torque and do not attain maximum torque until near synchronous speed is reached. This presents a problem in driving control system loads which require maximum torque at start-up and reduced torque as speed increases. The speed-torque characteristics of the motor can be varied to some degree by construction and materials used in the motor and armature, however, matching of speed-torque characteristics to load requirements is more efficiently accomplished by motor control circuitry.

Advancements in solid state technology have made available components for motor control schemes which greatly enhance control of AC induction motors and their application in control systems. The thyristor and Power Darlington transistors are key elements in such control circuits. The following control methods were studied (044, 159) in the Space Shuttle program for primary control:

a. Pulse modulated system with constant frequency (400 HZ)

b. Variable power/frequency

c. Bang-bang control with and without no-backs.

The pulse modulated system was rejected because of inefficiency. Motor losses due to heating were high because motors were operating at or near stall and continuously being pulsed with high starting current.

Reference (044) describes a method of developing variable power/frequency for application to AC motor. This design offers advantages in that a low frequency and voltage are applied to the motor giving better starting characteristics. Essentially, thyristors are used in a switching arrangement and pulsed "ON" at selected intervals to pass those portions of the basic power system waveform required to synthesize lower frequency waveform.

As described in Reference (159), the bang-bang control system motor is switched "ON" when a new surface position is commanded and switched "OFF" as the surface approaches the commanded position. This gives better efficiency than the pulse modulated system.

The study of Reference (159) determined that a combination of variable power/ frequency and bang-bang control with no-backs offered the most advantages. Development effort is required to implement such a system, and it was this development and lead time that caused the study of Reference (159) to recommend use of hydraulic rather than electro-mechanical actuators.

Technology and components are presently available for implementing the motor control system and electro-mechanical actuator. Since the concept offers possible advantages in future systems, it is considered worthy of development and testing in a flight control application to obtain experience on an operating system.

3.3.3 COMPUTER/ACTUATION INTERFACE. The actuation system has the primary requirement of delivering relatively large amounts of controlled power to aircraft control surfaces in response to commands generated by a central processor. The actuation system is required to develop several horsepower and this can be accomplished by various electro-hydro-mechanical methods. However, since digital computers typically supply only a few milliwatts at most, some form of electrical power gain or power conversion under control of the computer is required. Furthermore, since computer output is inherently digital and actuation system output is basically analog, some form of digital to analog conversion is required.

The traditional approach is to convert the digital command into an analog voltage or current electronically, and apply the analog signal to the load.

A second approach is to use a power controller more responsive to digital signals, but which still provides electric power output in analog form. This mechanization is typified by systems where an inverter uses SCR bridge rectifiers under computer control to vary the frequency and voltage of the power form applied to synchronous or induction ac motors. Very precise control of large amounts of electrical power has been achieved in this way.

A third approach is to use a power device that responds directly to digital commands. In precision, open loop systems, this approach is typified by stepping motors having 4, 5 or even 8 windings. A high power, high gain solid state switch is required for each winding, actuated by discrete pulsed commands from the central controller.

Another example is the hybrid digital servo actuator. In this case, a PM motor having 2^n windings is used. Each winding is weighted in terms of ampere-turns. Thus, the D/A conversion can be an inherent part of the power conversion unit.

One further method of integrating D/A conversion into the actuator is the "time-dwell" or pulse-width modulated drive scheme. For this case, the conventional actuator valve coil is fed a pulsed waveform whose on/off time is indicative of error signal, at a pulse rate higher than actuator bandwidth. The actuator acts as an integrator, and assumes a position equal to the average value of the driving waveform.

To close the loop, digital feedback signals from various elements of the actuation system are fed back to the central processor. These include position and possibly rate signals for closing the servo loop as well as signals needed for system monitoring.

The "JO block" actuator is another example of using the actuator to accomplish the D/A conversion.

Various methods of performing signal conversion electronically are reviewed in the following paragraphs. Conversion equipment is commonly classified as A/D (analog to digital), D/A (digital to analog) and D/D (digital to digital). While the equipment used is similar, each application is slightly different.

3.3.3.1 Digital to Digital Conversion. Data transfer in most computers is handled in a parallel operation. Therefore, any computer outputs of a serial, discrete, or incremented type require D/D conversions. Other forms of D/D conversion include changing the time base or change of logic (voltage) levels. The first two types, i.e., changing the data form or changing the time base, require temporary storage or buffering. The last form, changing the voltage level, can be accomplished within the line drivers. This is a simple change from the circuit standpoint; however, if large voltage changes (10 volts or greater) or large output currents are required, it becomes necessary to use discrete components which will increase size and cost.

Changing of data forms, while more complex from a circuit standpoint, can usually be accomplished with standard digital circuitry. The storage registers can be simple flat-packs. Logic circuits can also be flat-packs or integrated circuits. Because of the reliability of D/D equipment, these conversions are usually overlooked; however, this is the most common type of conversion. Whenever two digital machines work together, a conversion must be made -- at least time buffering, and often a conversion of the signal form. While these conversions are performed with standard digital hardware, each type usually requires special design effort.

3.3.3.2 D/A Conversion. A D/A converter (181) is a device that generates an analog voltage or current proportional to the value of the digital-input word. Basically, it consists of a stable reference (can be added externally), a set of binary weighted switches, and a precision resistor ladder network. An input register and output op amp are sometimes included.

The most fundamental D/A conversion scheme is shown in Figure 3.3-9. The input digital logic operates the switches, each contributing an appropriate amount of output current in binary weighted increments from a most significant bit (MSB) to a least significant bit (LSB), from left to right. The op amp converts the negative current to an output voltage. This approach is very simple and low in cost, but can only be used for medium-to-low resolution (10 bits or less) converters due to ladder network limitations.

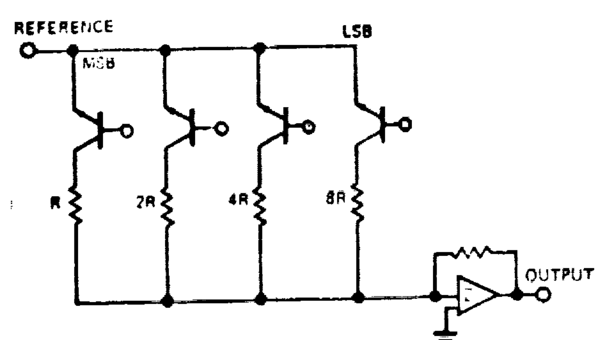


FIGURE 3.3-9 BASIC D/A CONVERSION SCHEME

One way to avoid resistance ladder limitations is to use the binary resistance quads shown in Figure 3.3-10. Each quad is essentially a 4-bit D/A converter with four different resistance values of R, 2R, 4R and 8R. By connecting two such quads along with an attenuating resistor in the current summing bus, an 8-bit unit is produced. The attenuating resistor used here reduces the LSB quad factor to the MSB quad by $2^4 = 16$. As more quads are added on, resolution is increased. One advantage of this method is that only the MSB quad resistors have the highest tolerance. Each successive quad can use resistors whose tolerance can be slightly less than the previous quad stage.

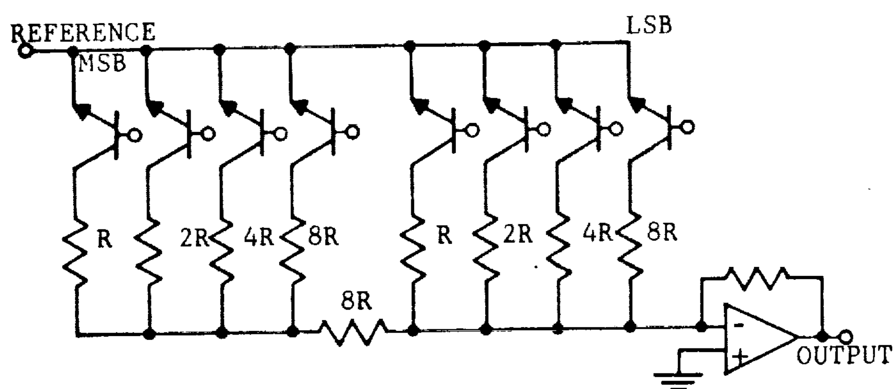


FIGURE 3.3-10 D/A CONVERTER USING RESISTANCE QUADS

The most popular method, derived from the resistance quad method, is the R-2R ladder (Figure 3.3-11). This method uses only two resistor values per bit in an R-2R relationship. The resistors must have close tolerances, however. This method is by far the most widely used.

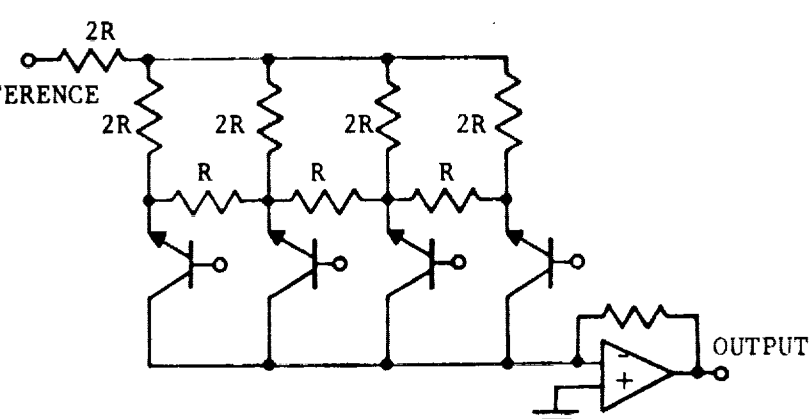


FIGURE 3.3-11 D/A CONVERSION USING R-2R LADDER NETWORK

Basically, D/A converters are available with either a fixed internal (or external) reference or with an external variable reference (multiplying types). Fixed-reference D/A converters that do not include an internal reference source allow the user to utilize a more stable reference than he would otherwise obtain with a converter having its own built-in reference. In general, fixed reference converters are more accurate than multiplying types because of the variable nature of the latter's reference.

Multiplying D/A converters produce outputs that are directly proportional to the product of the digital input multiplied by a variable analog reference.

Functionally, D/A converters are available as current-output or voltage-output types. The former do not include output amplifiers and are not restricted by their bandwidth limitations. Settling times well under 1 μsec (as low as 25 nsec) can be obtained with output currents of approximately 10 mA or less (output voltage ranges from 1 to 2V).

Because the output amplifier is not included, current D/A converters tend to be a little less expensive than voltage types. However, they may not be as temperature stable since the output is directly influenced by the temperature stability of the converter's resistance ladder network. Their applications are in areas where speed is paramount.

Voltage-output converters have output amplifiers. Because of this, their settling times tend to be above the 1 μsec range. However, a high-performance and high-speed output amplifier can be added to a high-speed current-output converter to make the output voltage a little faster in settling time, although this may cost more.

3.3.3.3 A/D Conversion. Fundamentally, A/D converters (181) either convert the input analog signal (either voltage or current) to a frequency or a set of pulses whose time is measured to provide a representative digital output or compare the input signal with a variable reference using an internal D/A converter to obtain the digital output.

Voltage-to-frequency, ramp, and integrating-ramp methods are the three leading conversion processes that use a time-measurement principle. Successive approximation and parallel/modified parallel circuits rely on comparison methods.

Figure 3.3-12 shows a typical voltage-to-frequency converter. Here, the input analog signal is integrated and fed to a comparator. When the comparator changes its state, the integrator is reset and the process repeats itself. The counter counts the number of integration cycles for a given time to provide a digital output.

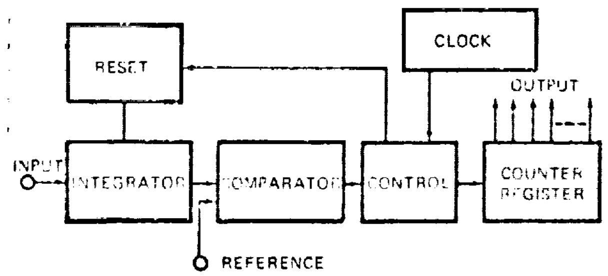


FIGURE 3.3-12 A/D CONVERSION

The principal advantage of this type conversion is its excellent noise rejection due to the fact that the digital output represents the average value of the input signal. Voltage-to-frequency conversion, however, is relatively slow because it operates bit-serially (approximately 1000 conversion/sec max.).

Ramp conversion works by continuously comparing a linear reference ramp signal with the input signal (which is converted to a pulse) using a comparator (Figure 3.3-13). The comparator initiates a counter when changing state, which counts proportionally the time the comparator is logically HIGH; the time itself being proportional to the magnitude of the input signal. The counter provides the digital representation of the input.

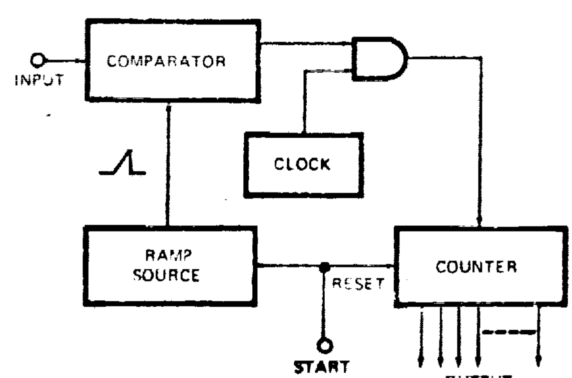


FIGURE 3.3-13 RAMP METHOD FOR A/D CONVERSION

This method is slightly faster than the previous one, but it requires a highly linear ramp source in order to be effective. It does offer good 8-to 12-bit differential linearity for applications requiring high accuracy.

With the integrating ramp converter (Figure 3.3-14) or better known as the popular dual-slope converter, the input analog signal is integrated over a fixed period of time followed by the integration of a fixed reference voltage of opposite polarity bringing the output of the integrator network to zero. Since the time it takes to integrate the reference voltage is proportional to the magnitude of the input signal, measuring this integration time with a counter and a pulse source results in an accurate digital representation of the input signal.

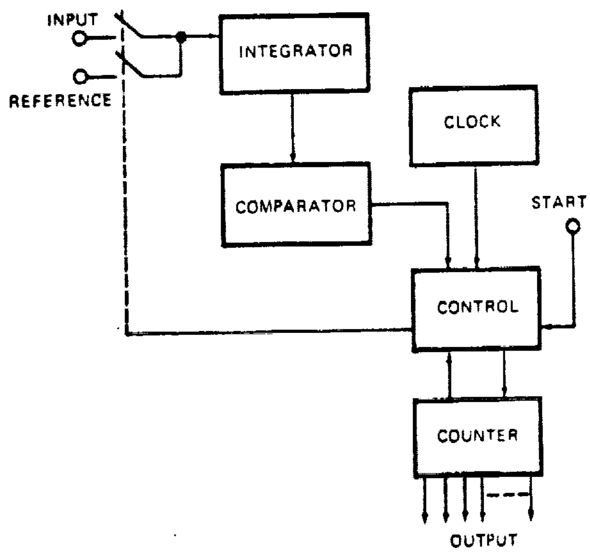


FIGURE 3.3-14 INTEGRATING RAMP A/D CONVERSION

While a relatively slow process, integrating ramp conversion offers high noise rejection and excellent stability with both time and temperature. It can be modified to increase its conversion speeds of approximately 2000 conversions/sec to more than 10,000 conversions/sec.

The simplest type of A/D converter is the parallel type (Figure 3.3-15). It uses one comparator for each input quantization level (i.e., a 6-level converter would have 6 comparators, and an 8-level unit, 8 comparators). Conversion is straightforward; all that is required besides the comparators is logic for decoding the comparator outputs.

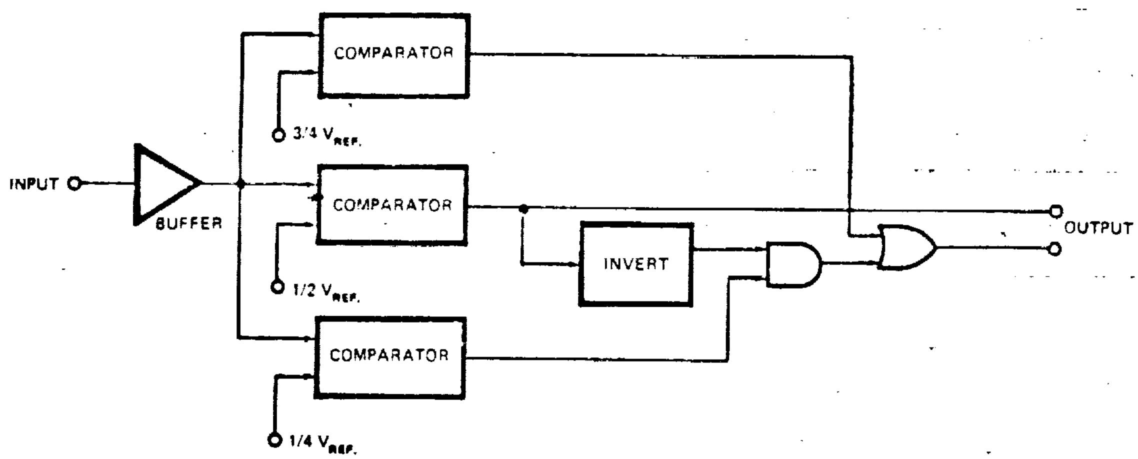


FIGURE 3.3-15 PARALLEL CONVERSION METHOD

Because only comparators and logic gates stand between the analog inputs and digital outputs, extremely high speeds of up to 50,000,000 samples/sec can be obtained at low resolutions of 6 bits or less. The fact that the number of comparator and logic elements increases with resolution obviously makes this converter impractical for resolutions greater than 6 bits.

A successive approximation A/D converter is shown in Figure 3.3-16. This encoding technique is the most popular of all A/D conversion methods because it offers a favorable combination of simplicity and relatively high speed. Also a direct serial digital output can be provided if single channel transmission is desired.

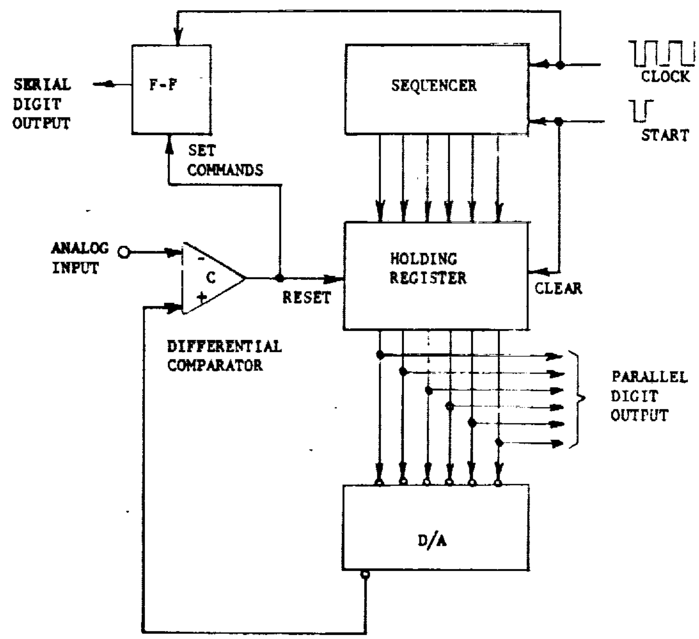


FIGURE 3.3-16 SUCCESSIVE APPROXIMATION A/D CONV.

3.3.3.4 <u>Discussion and Conclusions</u>. Digital-to-digital conversions are common and can be performed with standard digital components, if it is merely desired to shift logic levels or interface one logic family with another. If it is necessary to drive a digital device requiring greater electrical power than a few milliwatts, then high-power solid state switches such as power transistors, SCR's, or opto-isolators would be needed and the conversion is no longer in the strictly digital-to-digital regime.

The ideal electronic interface is one where the actuator requires at most only a few milliwatts of pulsed digital energy. Such actuators are presently undergoing laboratory tests, but actuator reliability has not been established and much development is needed.

A/D and D/A converter technology has advanced to the stage where monolithic converters are available. Precision Monolithics has a 10-bit D/A converter that is self-contained within a single chip. This company has recently produced a monolithic 12-bit A/D in a 40-pin DIP, but its temperature range is limited to 70°C maximum. However, they expect to have a military-grade unit available in the near future.

Actuator design need not be compromised merely to delete the need for a D/A unit, since the conversion is only a fraction of total system weight. Thus, "analog actuator" technology which has undergone considerable development in recent years can be used to advantage in realizing a reliable fly-by-wire system.

3.3.4 SENSORS

3.3.4.1 Typical Sensors. Closed loop actuation systems require actuator position and/or rate feedback signals. Position transducers suitable for aerospace applications are presently limited to potentiometers, synchros, LVDT's and so-called digital encoders.

Because potentiometers are contacting type devices, they have a much shorter life than synchros or LVDT's. Therefore, potentiometric contacting transducers are not given serious consideration for use in FBW systems.

Synchros are the simplest and most reliable of present day position sensors. This device has only two windings, and can be constructed without the need for slip-rings or brushes by limiting shaft rotation to ± 1.57 rad ($\pm 90°$). Synchros are well suited to angular position sensing and in flap angle or rotary actuator position sensing.

LVDT's are by far the most popular class of linear position transducers. Their rectilinear mechanical design makes them directly compatible with linear actuators without the need for rotary-to-linear conversion. Like synchros, LVDT's are contactless and frictionless. The life expectancy of LVDT's is slightly less than that of synchros. The difference in reliability is minor, as indicated by the fact that LVDT's can be built with multiple-redundant windings for use in high reliability applications. Packaged units are also available with built-in electronics to generate AC excitation and perform the demodulation function; these are called DC LVDT's. It is reasonable to assume that LVDT's could be constructed with built-in analog-to-digital conversion circuitry. The desirability of such a "digital LVDT" even in an all-digital FBW system remains to be proven, however.

Digital encoders are available from several companies. Many models have been qualified to military environmental requirements. All of these models are rotary encoders. Most premium encoders use the photolectric principle (electro-optical transducers). This design requires no brushes and the result is a "contactless" encoder having (presumably) better reliability than brush-type encoders originally used for rotary shaft encoding.

Linear digital encoders are available for commercial and industrial applications. These units use either photo-electric or magnetic effects. In the photo-electric models, a transparent shaft is masked with the desired digital code, and an LED is used as the light source. Photo-transistors are commonly used as sensors to perform the read-function; photo-diodes could also be used. In units using the magnetic principle, the sensing shaft is encoded with a magnetic strip containing the desired code. A "read-head" is used to provide direct digital readout.

Linear encoders could be developed to meet the resolution requirements. The technology is within the state-of-the-art. Most present designs are ferrite core read-heads of the type used in computer memories. These require a high frequency signal for interrogation. The rod is made from a high coercive force magnetic material, and contains a unique flux pattern for the desired code. Hall Effect devices could possibly simplify the interrogation electronics; however, the technology is not sufficiently advanced.

A linear photo-electric digital encoder could be developed to withstand military environments; however, mechanical packaging would have to be arranged to keep the unit free internally of hydraulic fluid. As a result, package size could become excessive, compared to a design using LVDT's which can be immersed in hydraulic fluid.

3.3.4.2 Sensor Selection. If a linear digital encoder were available, its place in a FBW system must be evaluated. First, would coding be incremental or absolute? In an absolute device, the output signal corresponds to a unique position on the scale. One advantage of absolute devices is that loss of power, such as a momentary outage or overnight shutdown, does not cause position information to be lost. On the other hand, incremental systems produce a periodic signal corresponding to the periodic nature of the encoded scale. The whole number traversed must be counted and remembered; therefore, a loss of power causes position information to be lost. This allows the zero position to be easily set anywhere within the range of travel, but would require a nulling process after each application of power.

If the computer is "inside" the actuator servo loop, and a position servo is desired, then absolute encoding would be preferred. On the other hand, if a pulse-width-modulated servo is used, then rate feedback only might be preferred for loop stability. In this case, incremental encoders could be used.

At the present time, LVDT's are preferred as actuator position feedback transducers. If a digital feedback signal format is required, then a commutating A/D in the computer's I/O could accommodate the demodulated analog output of several LVDT's, so that only one A/D would be needed per redundant channel. With size reductions becoming possible through monolithic techniques, individual A/D's for each LVDT might be feasible.

If the computer is "outside" the actuator loop and an analog actuator is used, then LVDT's would still be preferred. Additional conversion could be performed on the demodulated LVDT ac output to provide digital data to the computer for servo monitoring and failure detection purposes.

3.3.5 REDUNDANCY MECHANIZATIONS. It is a generally accepted premise that a single channel FBW system is unacceptable from reliability and safety aspects. Redundancy in some form is necessary to instill confidence and raise the loss rate reliability to an acceptable level. Highly reliable components are beneficial to overall reliability but this alone does not solve the problem, especially in the actuation area where exhaustive test data can not be accumulated to prove predicted reliability. Redundant flight controls have been the subject of intensive study for the past two decades as attested by the fact that nearly 80% of the material found in the literature search was concerned with the design of redundant control systems. The redundancy tree (142) shown in Figure 3.3-17 depicts various approaches to redundancy. As can be seen, a large number of approaches are possible. The number of specific designs is multiplied a thousand fold when considering the various components and combinations available to mechanize any one approach. It is no small task to review, evaluate and draw summarial conclusions from all the preceding effort. The Air Force has recently contracted for a review of all study effort to date, regarding redundant FBW control systems.

Considering the risks associated with generalizations and over simplification, the following conclusions summarize the effort to date:

1. A number of approaches are available which can satisfy the overall loss rate reliability requirements. Selection of the best approach in each specific design is determined by requirements other than reliability.

2. Active redundancy is generally preferred to stand-by redundancy because; (a) failure transients are less severe, (b) system operation is not predicated upon positive action in the event of failure (in stand-by redundancy the failed channel must be immediately removed--thus a series active element is required), and (c) monitoring requirements are less severe and permit greater flexibility in system design.

3. Redundancy alone is insufficient to assure loss rate reliability; some method of assuring that all redundant channels are functional at periodic internals is required.

4. Monitoring is an efficient method of improving operational reliability.

In addition to the preceding conclusions drawn from the literature, the following philosophical views, observations, and opinions regarding the design of FBW are offered for consideration:

a. Flight control systems need to be as simple, direct and foolproof as possible with regard to design, operation, inspection and maintenance.

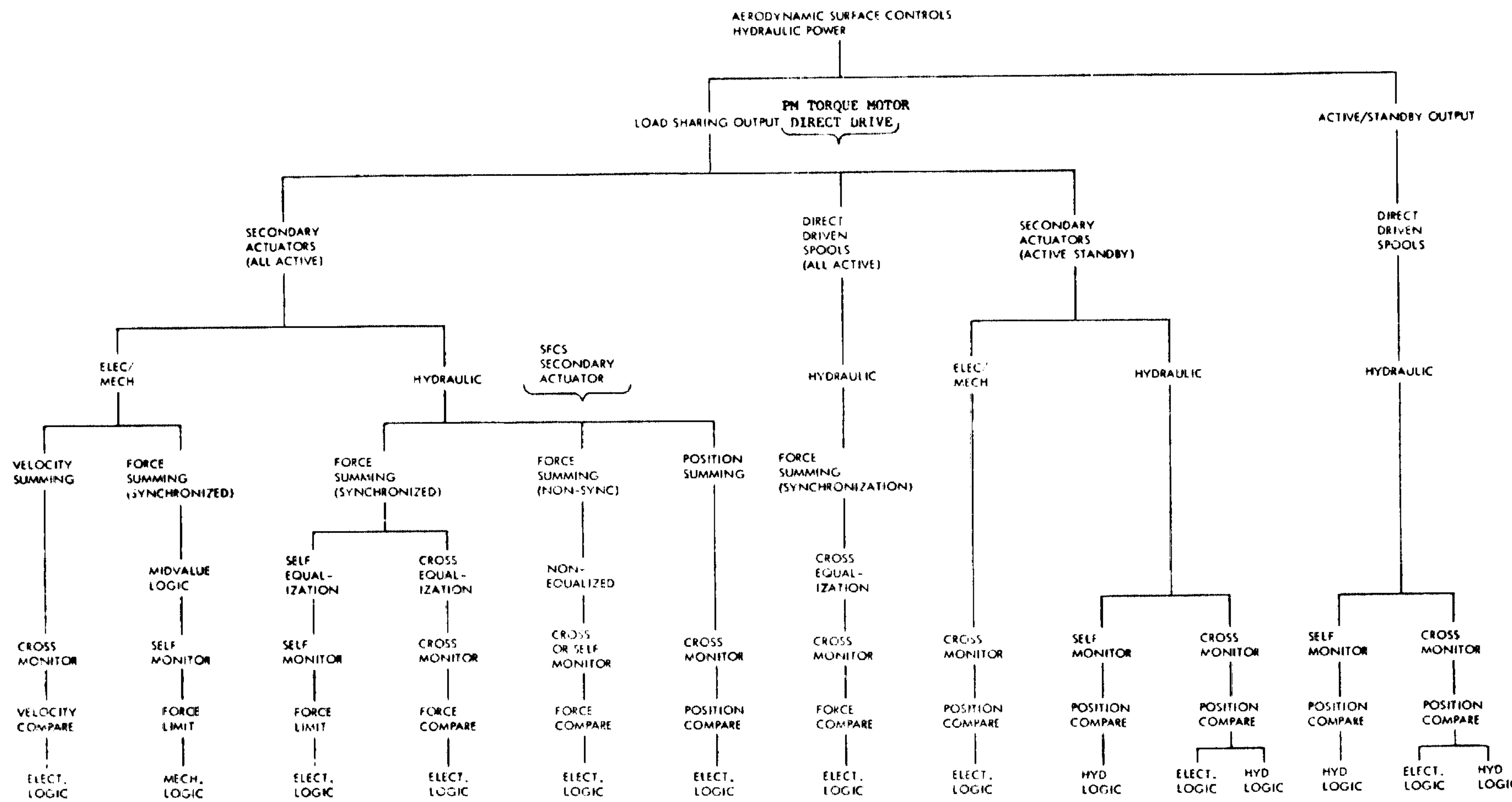


FIGURE 3.3-17 REDUNDANCY TREE

b. There are two methods of achieving the required level of reliability: (1) develop extremely reliable parts; or (2) design reliable systems utilizing redundancy techniques. Neither approach in itself is a totally acceptable solution. Development and verification of components having the necessary reliability is a long term project while indiscriminant system redundancy leads to impractical complexity and unwieldy packages. A judicious blend of the two approaches is necessary.

c. The first step of fundamental importance toward achieving high reliability in FBW control systems is minimizing the number of components, especially small delicate parts. The second step is to design in reliability through "brute force overdesign",(i.e., design margins in excess of maximum expected values).

d. Systems should be designed to minimize maintenance actions, thereby precluding "Murphy Failures" or human errors. The only reliability number that counts is the one achieved in the field where maintenance is often performed under less than ideal conditions. The way to achieve field reliability is to design extremely rugged equipment which can take abuse and misuse and which requires minimum attention to avoid "user induced" failures.

e. System reliability should be obtained through maximum use of high reliability subsystem elements and minimum redundancy.

f. Anticipated failures are rarely problems; these are generally accounted for in design. It is the unanticipated failures that are catastrophic. Non-identical system redundancy has considerable merit especially for operation in an unknown or poorly defined environment. The alternative is sufficient testing.

g. Tremedous advances have been made in the reliability of electronic circuitry with the advent of solid state components. Integrated circuit techniques permit manufacture of complex circuits having reliability equaling a single transistor. Now LSI techniques appear capable of producing entire systems having comparable reliability figures. The reliability of hydro-mechanical and electro-mechanical transducer components have advanced only modestly during the same time period. The electronic circuit is now the reliable element, easily made redundant, and electro-hydraulic and hydro-mechanical components are now the unreliable elements with greater penalties paid for redundancy. In view of the relative reliability, it is logical that electro-mechanical electro-hydraulic, hydraulic, and mechanical components where large penalties are paid for complexity, should be designed as simple and reliable as possible, and place the computational complexity in the electronics area where minimum penalties are paid.

h. A basic problem in the power actuation area is in determining the components which must be redundant, and selecting the best approach to either eliminate the need for redundancy or design the required redundant system. It has been suggested that the two fail operational criteria, so often used in the controls area, be carried through to the surface, (i.e., split surface). The following observations on the split surface approach to redundancy are offered for consideration. Study of aircraft losses due to failure of a surface indicates a loss rate in the 0.01 - .03 x 10^{-6} LPFH range. The main protection afforded by split surfaces is against actuator jamming or seizure. Jamming potential in hydraulic power actuators is remote. Review of failure rate data (Appendix 3-1) indicates an overall seizure failure rate of 0.08 x 10^{-6} FPFH, in the same neighborhood as a surface failure.

Splitting surfaces has limited effectiveness in protecting against free surface failures caused by a break in the surface actuator. If the surface does not assume a trailing position the protection is questionable. The penalties of splitting some control surfaces in high performance type aircraft is prohibitive.

There are other approaches to protecting against free surface type failures such as more structurally sound actuators (114), multiple actuators per surface, and better utilization of the inherent redundancy afforded by existing controls. The splitting of some control surfaces on high performance aircraft could not be accomplished without prohibitive penalties. The advantages to be gained from surface redundancy should be investigated thoroughly before they are accepted.

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3.4 TRADE STUDY

3.4.1 SELECTED CONFIGURATION. There are a large number of approaches for implementation of fly-by-wire systems. They all have merit in some way--at least in the mind of the investigator who expended considerable time and energy pursuing the concept. It can also be concluded that there are a number of approaches available which are capable of satisfying the minimum requirements. The problem facing the designer is selecting the most suitable or optimum approach. Attempts have been made to apply the power of the computer in optimizing the overall system (008). The optimum or most appropriate of highly competitive approaches becomes very dependent upon the criteria utilized in defining optimum.

Emphasis on past designs has been primarily on mission reliability with performance a close second; maintainability, cost and weight are generally rated far less important. This has resulted in two-fail operational requirements or essentially a four channel system with complex, costly, and hard to maintain equipment whose mission reliability is somewhat questionable. Experience indicates that the most field-reliable equipment is equipment that requires the least maintenance. Equipment which requires continuous maintenance suffers accordingly from the human element. The more a system must be touched, the more problems are created. Ultimate safety is not necessarily improved by redundancy; added reliability achieved through redundancy can easily be lost in the monitor and ultimately be lost in field maintenance. It is believed that maintainability should be considered equally important with reliability and performance. This leads toward the simple brute force approach to built-in reliability. With this in mind, the following approaches were selected for more detailed study. These approaches represent what are believed to be the most competitive and basically different mechanizations.

(1) PM Torque Motor Direct Drive of the Power Actuator Spool/Sleeve Valve
(2) Stepper Motor Direct Drive of the Power Actuator Spool/Sleeve Valve
(3) Force Summed Secondary Actuator Driving Power Actuator Spool/Sleeve Valve
(4) Electro Mechanical Actuation

Table 3.4-1 compiles parameters on these four concepts. The following paragraphs describe the four concepts and present an evaluation and discussion of the parameters considered most important in selecting the actuation system for the digital fly-by-wire program.

TABLE 3.4-1 CONFIGURATION DESCRIPTIONS

SYSTEM / PARAMETER	FORCE SUMMED SECONDARY ACTUATOR	STEPPER MOTOR DIRECT DRIVE	PM TORQUE MOTOR DIRECT DRIVE	ELECTRO-MECHANICAL
CONTROL	Position	Position	Velocity	Velocity
INPUT SIGNAL	Analog	Digital	Analog	Digital
OUTPUT	Analog	Analog	Analog	Analog
FEEDBACK	Elect/Mech Analog	Mechanical Analog	Electrical Analog	Electrical Analog
POWER SOURCE	Hydr Pump	Hydr Pump	Hydr Pump	ac Generator
DRIVE TYPE	Variable Displacement	Variable Displacement	Variable Displacement	2000 Hertz
CONTROL METHOD	Spool/Sleeve	Spool/Sleeve	Spool/Sleeve	Variable Power & Freq.
ASSOCIATED HARDWARE	Figure 3.4-3	Figure 3.4-2	Figure 3.4-1	Figure 3.4-4
OPERATION RANGE	Variable & Adequate	Variable & Adequate	Variable & Adequate	Variable & Adequate
DESIGN SCALE-ABILITY	Adequate	Adequate	Adequate	Adequate
RATE	Variable & Adequate	Variable & Adequate	Variable & Adequate	Variable & Adequate
BACKLASH	Adequate	Adequate	Adequate	Adequate
RESOLUTION	Adequate	Adequate	Adequate	Adequate
TRIM	Adequate	Adequate	Adequate	Adequate
DESIGN CONCEPT SUITABILITY	Adequate	Adequate	Adequate	Limited

3.4.1.1 PM Torque Motor Direct Drive. The first two approaches are in accord with system design philosophy expounded throughout this report. Design begins by selecting the simplest, ruggedest, most reliable power element available -- the linear hydraulic actuator and spool/sleeve valve. In the PM torque motor approach, Figure 3.4-1, high strength torque motors are coupled directly to dual tandem valves. Each torque motor has four coils driven by four amplifiers. The amplifiers are connected in an adaptive gain arrangement whereby overall gain changes resulting from failure of an individual amplifier are self-compensated. This minimizes performance degradation inherent in force summing redundancy. The four amplifier outputs are force summed in the torque motor and are further force summed with the four amplifier outputs driving the other torque motor. Mechanical parts are minimized to only a few large, rugged, mechanically over-designed components, in a brute force approach to reliability. In the electronic area, massive redundancy is employed to minimize failure effects. This is accomplished with minimum cost, weight, and power penalties. Monitoring is eliminated. In force summing, a choice is usually available between providing monitoring or additional redundancy to improve reliability. Additional redundancy is selected for reasons of simplicity and submersion of failure effects. This PM torque motor approach is in keeping with the philosophy that small, hand-build, parts should be avoided. The bulk of the design complexity is placed in the electronics area where parts can be essentially produced by machine, thereby minimizing the human element and permitting exhaustive testing of the complex portion.

Two methods of feedback are possible: mechanical feedback where a force proporational to actuator position is summed with the torque motor forces, or electrical feedback utilizing a redundant LVDT buried inside the actuator for protection. Mechanical feedback in FBW systems has been used extensively in missile and space vehicles. LVDT's have become a standard electrical feedback element for aircraft flight controls. The survey disclosed no sensors more suitable for this application. A clear choice between mechanical and electrical feedback is not readily apparent; further study is required. The choice should be made on basis of reliability, maintainability, and other specifics associated with a particular application. Electrical feedback is selected for the PM torque motor direct drive approach in contrast to mechanical feedback selected for the stepper motor direct drive approach.

The direct drive approach holds promise of a number of advantages -- in cost, weight, simplicity, maintainability, and inspection requirements. Flight controls should be as simple, direct, and foolproof as possible. Therefore the ultimate FBW control system is the simplest; a dual channel system constructed from sufficiently reliable components wherein one channel with sufficient reliability is backed up by a second channel to take care of the human aspect. The crucial element in this approach is the control valve which must be better than current devices. Technology for more suitable valves exists today; it only remains to be proven.

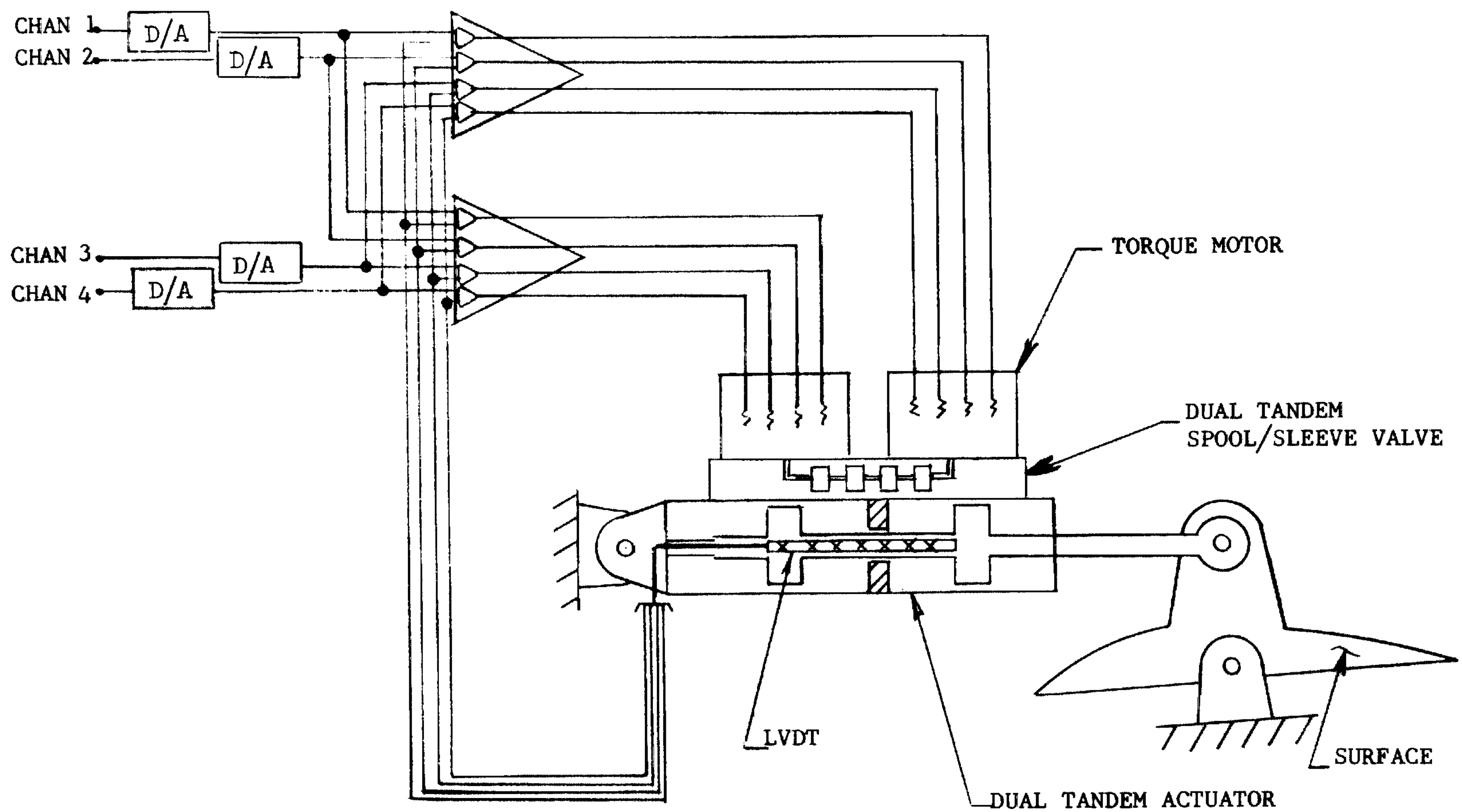

FIGURE 3.4-1 PM TORQUE MOTOR DIRECT DRIVE CONCEPT

3.4.1.2 Stepper Motor Direct Drive. In the stepper motor direct drive approach, Figure 3.4-2, dual stepper motors are position summed, through a differential mechanism, with a mechanical feedback signal. Mechanical feedback is obtained by coverting linear motion to rotary motion by a ball screw helix integrated within the piston rod. Low efficiency gearing or braking is used to prevent back-driving the differential mechanism should a motor become unpowered. The stepper motors function to open the valve while mechanical feedback functions to close the valve. Thus, high force levels via the feedback mechanism are available to shear out contaminants. The stepper motor output must overcome friction, flow forces, and valve silting. The valve-motor feedback assembly is integrated into the actuator housing where it is protected and bathed in oil. This feedback design permits grounded body operation. In essence the actuation system is a dual parallel electrical stepper motor with a linear hydraulic actuator to provide muscle. Accuracy is dependent upon the stepper motor. The stepper motor arrangement is inherently fail passive and inherently redundant. Typically the motor has five phases each with a phase coil which could be split for redundancy. A short or open coil produces no response -- only degraded performance. Rotations or stepping requires specific sequencing of the phase coils. Applying continuous voltage or denying voltage will not produce runaway. The necessary sequence of signals required to cause runaway makes this possibility very unlikely. Digital to Digital (D/D) conversion is required to convert computer words into stepping signals.

3.4.1.3 Secondary Actuator. The force summed secondary actuator approach has been explored in several recent programs: NASA F-8 FBW BCS, Air Force 680J program, Space Shuttle, etc. This is undoubtedly the most thoroughly studied, developed and tested of all the approaches and is well documented in the literature (005, 029, 032, 142). Basically, the output of four small electro-hydraulic servos are mechanically force summed as depicted in Figure 3.4-3. The summed output is mechanically coupled to and drives a dual tandem spool/sleeve valve and actuator. Feedback at the power actuator is mechanical. Electrical feedback about each electro-hydraulic servo is provided by LVDT's. ΔP, a measure of force flight, is used for monitoring. The monitor senses a failed channel by comparing ΔP's between channels and if a channel ΔP differs from other channels by a specific amount for a certain period of time, a failure is assumed and the channel is de-energized. This approach was selected primarily as a baseline for comparison.

The force summed secondary actuator approach, such as depicted in Figure 3-4-3 is a viable design approach capable of meeting mission reliability requirements. Extensive detail design data, studies, test data, etc., exist and are readily available covering this approach. Unquestionably, a secondary actuator type system can be designed which can meet the requirements.

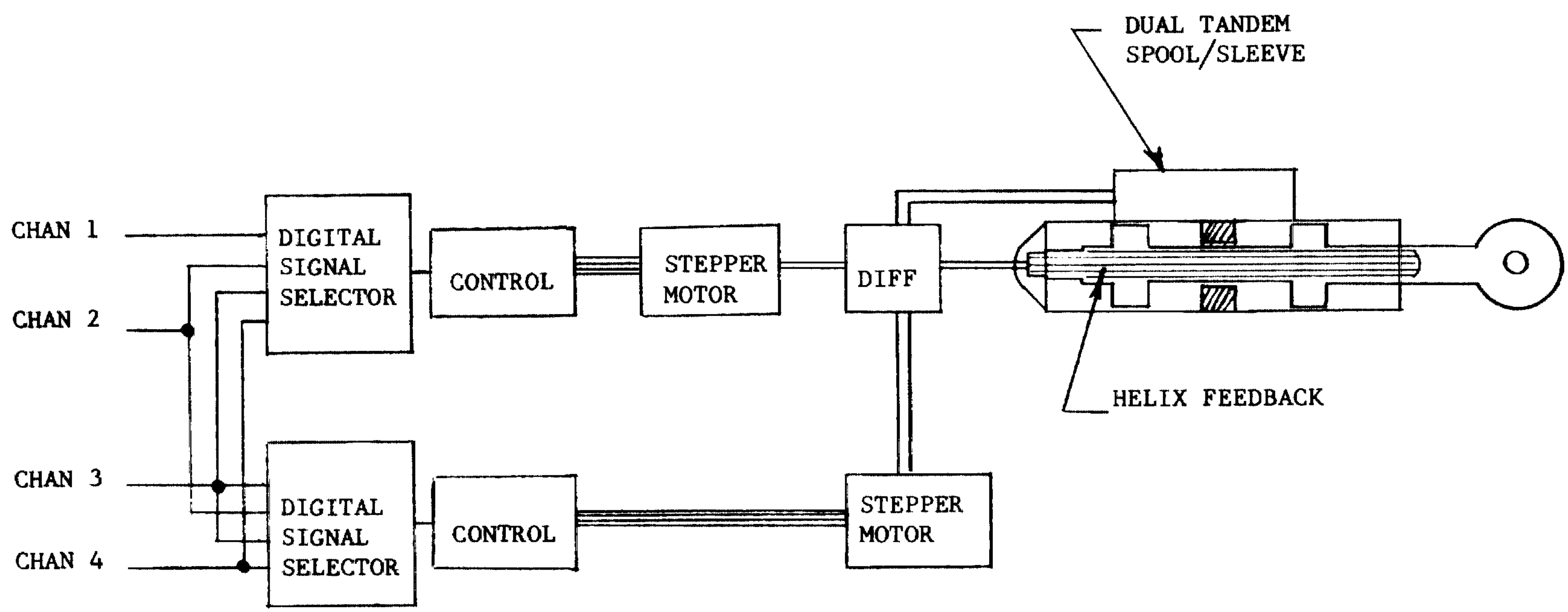

FIGURE 3.4-2 STEPPER MOTOR DIRECT DRIVE CONCEPT

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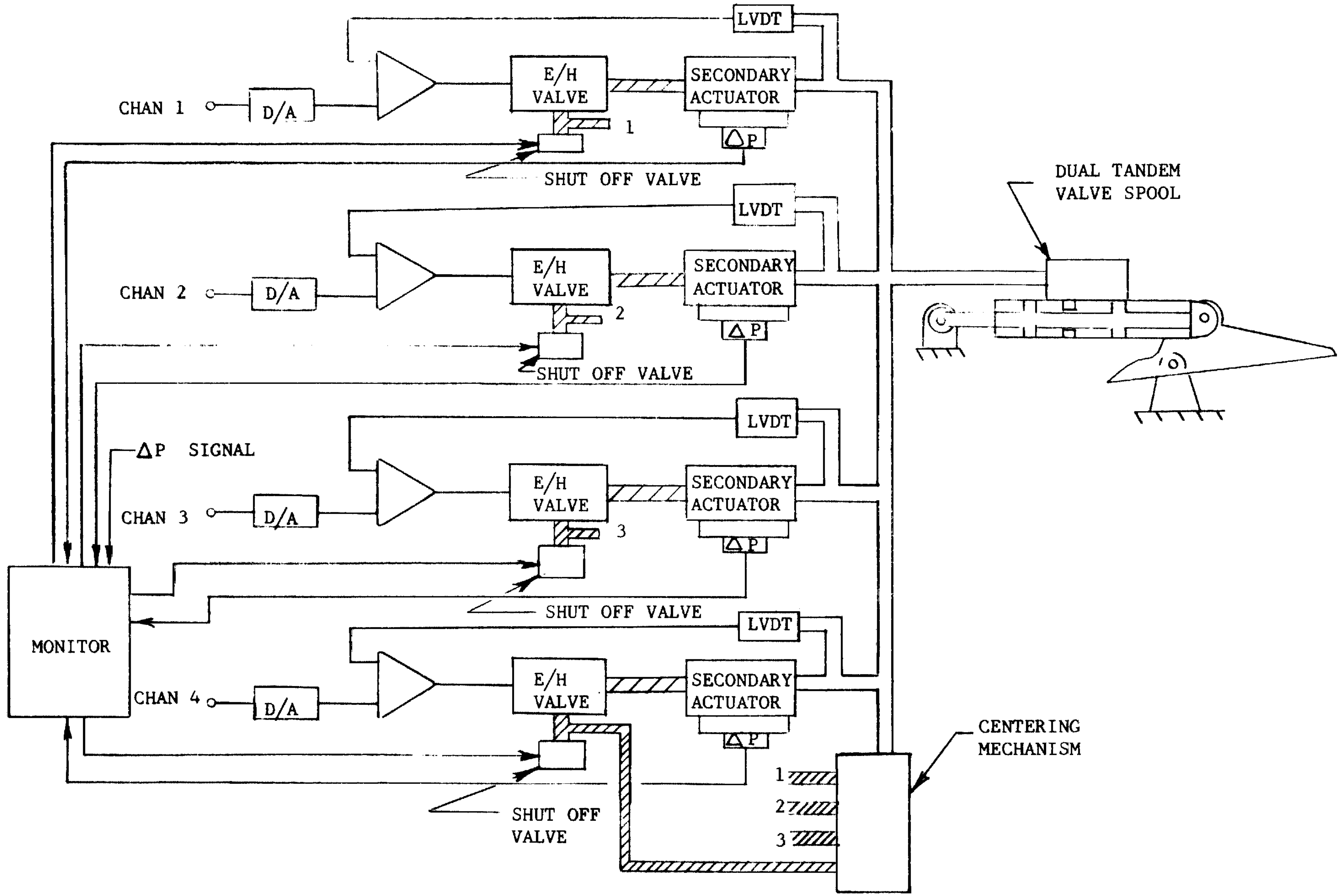


FIGURE 3.4-3 SECONDARY ACTUATOR

3.4.1.4 <u>Electro-Mechanical (EM)</u>. The electro-mechanical actuator concept is depicted in Figure 3.4-4. Input and feedback signals are fed to redundant signal selection circuitry which, according to mid-value logic, selects the command signals and transmits them to the motor controller. The concept provides for selection of the proper redundant input signals channel and provides a redundant source of mechanical power to drive the control surface. The solid state thyrister motor controller, a variable power/frequency bang-bang design, applies 3-phase power to the motors. Two motors are combined differentially to drive the nut and two to drive the screw. The motors are mounted to structure.

Electro-mechanical actuation holds promise of eliminating hydraulics, reduced fire hazard, elimination of contamination problems, easier power system fault detection and isolation, and simplified maintenance and repair. If the entire primary control function could be handled electromechanically, then hydraulics could conceivably be removed from aircraft, resulting in elimination of a whole class of problems associated with hydraulics -- leaks, fire hazards, contamination, etc. The state-of-the-art in EM, and particularly in electric motors and motor control appears to have advanced to the point of feasibility for primary flight controls at least for some classes of aircraft -- as the results of advances in materials and solid state power control devices.

ELECTRO-MECHANICAL ACTUATION SYSTEM

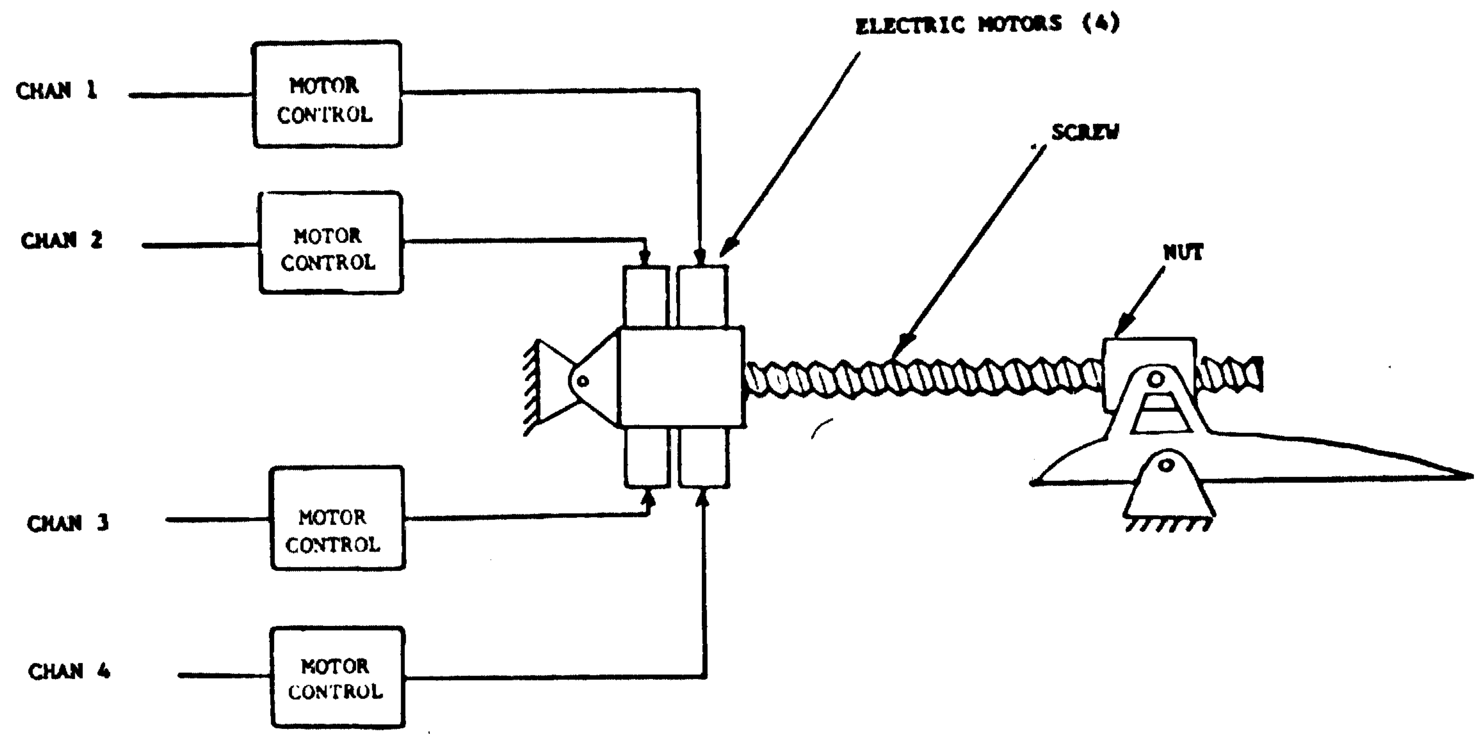


FIGURE 3.4-4 ELECTRO-MECHANICAL LINEAR ACTUATOR CONCEPT

3.4.2 DISCUSSION. Actuation system design involves a large number of factors. In most cases no single design is optimum in all areas, therefore trade-off between parameters is necessary. Factors or characteristics are considered most important in practical Fly-by-Wire Actuation System design are:

a. Loss Rate Reliability
b. Operational Reliability
c. Maintainability
d. Design Simplicity
e. Performance
f. Cost
g. Weight
h. Limiting Factors

Each configuration was qualitatively evaluated in relation to these factors. The comparison is summarized in Table 3.4-2. The PM torque motor direct drive approach rated highest; the EM approach rated lowest. This is not intended to mean that one approach is best with the exclusion of all others for all applications, rather it indicates a general preference for the applications presented herein.

TABLE 3.4-2 TRADE-OFF SUMMARY

TYPE OF SYSTEM / CHARACTERISTIC	SECONDARY ACTUATOR (HYDRAULIC)	STEPPER MOTOR DIRECT DRIVE (HYDRAULIC)	PM TORQUE MOTOR DIRECT DRIVE (HYDRAULIC)	ELECTRO-MECHANICAL
LOSS RATE RELIABILITY	GOOD	GOOD	EXCELLENT	?
OPERATIONAL RELIABILITY	FAIR	GOOD	EXCELLENT	FAIR
MAINTAINABILITY	FAIR	EXCELLENT	EXCELLENT	EXCELLENT
DESIGN SIMPLICITY	GOOD	GOOD	EXCELLENT	FAIR
PERFORMANCE	GOOD	GOOD	EXCELLENT	FAIR
COST	GOOD	GOOD	EXCELLENT	FAIR
WEIGHT	GOOD	GOOD	EXCELLENT	FAIR
LIMITING FACTORS	EXCELLENT	GOOD	FAIR	FAIR

3.4.2.1 Loss Rate Reliability. Loss rate reliability is defined as the probability on a per hour basis of not having a catastrophic failure resulting in aircraft loss. Three of the four approaches utilize a tandem spool/sleeve valve and tandem hydraulic actuator. These elements have an excellent record in regard to catastrophic failures (i.e., jam failure rate, .08 x 10^{-6} or break fail rate .01 x 10^{-6}). The significant difference between the three approaches in this area is the amount of force available at the spool to shear out contaminants. The secondary actuator is capable of producing nearly any force level desired. The direct drive stepper motor approach is capable of producing more than 2.22 k N (500 lbs) with the hydraulic helix two-stage valve. The PM torque motor is capable of the least force but develops a force as high as commonly used in mechanical control linkage designs which have performed satisfactorily for years. One method for redundancy against power spool jams is the dual concentric valve (i.e., a spool within a spool).

The PM torque motor approach provides dual mechanical load paths straight through to the valve spools. The secondary actuator and stepper motor approaches have a single mechanical linkage, but this is considered adequate providing it is sufficiently overdesigned. The stepper motor approach is complex, requiring gears, bearings, differentials, jack or ball screw helix, shafts, etc., all of which can be made reliable with proper design but not as reliable as the simpler approaches. The PM torque motor drive is accomplished by dual-quad (8) coils with individual electronic drive circuits. This design can function with three identical hard over type failures and as high as seven (7) non-identical failures. The secondary actuator drive is dependent upon quad (4) electro-hydraulic servos. This approach can function, with the aid of the monitor, after two failures and fails passive at the third failure. The stepper motor approach is predicated upon a fail passive design and inherent electrical redundancy within the stepper motor. One hard over short in the phase coil power switch degrades performance, two adjacent hard over shorts stall or lock the motor. The system is still operable via the other stepper motor channel which can withstand at least one power switching failure and is thus three fail operational. This approach is predicated upon design of a controller which is fail passive and takes advantage of the redundancy within the stepper motor.

The PM torque motor approach is the overall choice in regard to loss rate reliability. It has a higher level of redundancy up stream of the spool valve, is not dependent upon hydraulic components and is simpler when taking into account monitoring. The main question which can be raised is the lower spool drive force capability.

The EM concept employs a linear mechanical actuator. These actuators have an excellent loss rate reliability record; jam and break failure rates combined run about 0.16 per 10^6 hours. The mechanical actuator is driven by four electric motors -- two driving the jack screw and two driving the nut. Dual drive is considered adequate due to the basic reliability of the parts and the fact that only passive failure modes exist. The electric motors are three phase induction type and considered fairly reliable. The primary failure mode expected is bearings which have a failure rate of 10 per 10^6 hours. A simple motor brake is used for the no-back function. The motor controller is the critical element in this approach. Performance optimization suggests variable frequency, variable voltage with bang-bang control. This requires three phase power conversion in both frequency and voltage and on-off power switching which can be accomplished by solid state power switching devices. Electronic part reliability is very dependent upon temperature; high power levels result in high temperatures. The entire approach is dependent upon development of a highly reliable motor controller which should be fail passive if possible. There is little historical data in this area. Smaller EM actuators used in secondary controls have a failure rate of 4 to 5 per 10^6 hours, most of which can be attributed to electrical problems. Since secondary controls generally operate only a few times per flight, a more realistic failure rate for primary control applications would be perhaps 500 per 10^6 hours and for dual redundancy about 0.25×10^{-6} FPFH. The total system is not dependent on hydraulic power sources; this eliminates one large failure rate element in the total system. EM is more demanding on the electrical supply, however, which nullifies this advantage to some extent. Detailed total system studies covering the prime mover, power generation and distribution, are required for a comparison.

The essential conclusion reached in studies of EM actuation for primary Space Shuttle control application was that the EM approach is feasible and competitive but there would be considerable developmental risk in the motor controller design area.

3.4.2.2 Maintenance Reliability. Maintenance reliability is defined as the probability that all components are operating properly on a per hour basis.

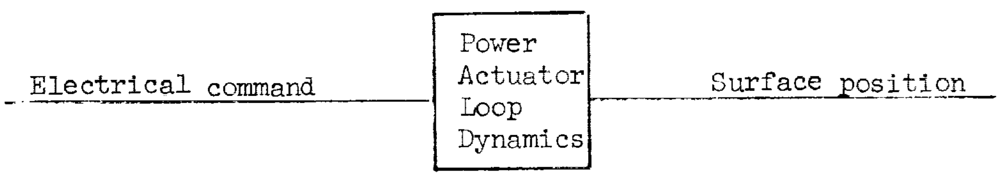
The PM torque motor approach is rated excellent simply because it has the fewest and simplest parts. The Stepper motor is rated above the secondary actuator since it has no multiple hydraulic servos. Complexity in the EM approach is primarily in the motor controller; this could be accomplished by LSI techniques. This area is expected to have a higher failure rate due to the number of high power switching devices.

3.4.2.3 Maintainability. The EM system is probably the easiest to maintain in the field because of elimination of the hydraulics. Failures would be isolated to a motor controller, a motor, or an actuator. Only the appropriate element would be removed The PM torque motor is rated good due to the fewer number of parts and high overall reliability. The stepper motor is rated fair and the secondary actuator is rated poor due to the number of hydromechanical parts. Failures in all three hydraulic approaches could be isolated to either the electronic or actuator package and removal and replacement of the appropriate unit for field maintenance.

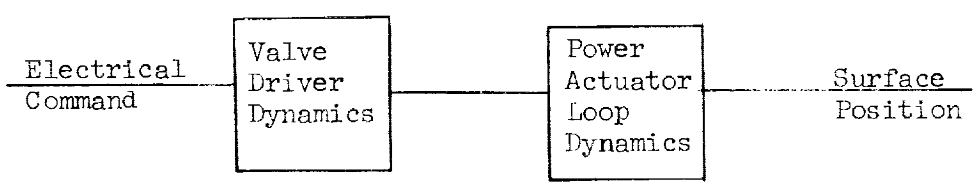
3.4.2.4 Design Simplicity. The simplest design is the PM torque motor approach; it has one hydraulic stage, two simple torque motors, an IC amplifier with a low power output stage, a quad LVDT and demodulator, and D/A converters. The PM torque motor approach is rated excellent. The stepper motor is rated good. It does not have the LVDT, demodulators, or D/A converters but does have a more complex D/D converter and mechanical feedback. The secondary actuator, compared to the PM torque motor design, is rated fair since it has four torque motors (although smaller) instead of two, four additional hydraulic servos, and a monitor. The EM approach is rated poor due to the number of individual high power electronic parts in the motor controller. Integrated circuitry is presently limited to low power levels, primarily milliwatts, although special IC's are becoming available which are capable of a few watts.

3.4.2.5 Performance. There are many parameters involved in defining performance of primary flight control actuation systems, each of which has a relative importance. These include force, stroke, rate, bandwidth, resolution, and backlash. The parameters for a given configuration are highly dependent on the individual design. The potential dynamic response is considered to be the best performance criteria which permits a trade-off of the given configuration without being dependent on the particular design of the configuration. The potential dynamic performance of the four selected configurations is evaluated in the following paragraphs.

The PM torque motor approach is capable of the highest dynamic performance, being limited only by power actuator/load dynamics. The PM torque motor valve driver has very high response (greater than 100 hertz), far beyond the required pass band. This approach is rated excellent. The secondary actuator and stepper motor approaches are next in response capability and are rated good. They are limited not only by the power actuator/load dynamics but also by the valve driver. This is the functional element which converts electrical command signals to mechanical valve command signals. A comparison of the two is given below:



PM Torque Motor Approach



Secondary Actuator & Stepper Motor Approach

The stepper motor and secondary actuator valve driver response can be made fairly high thereby minimizing but not eliminating their effect. To produce responses comparable to the PM torque motor approach, higher power actuator loop dynamics is required which can be accomplished only with large power and weight penalties. The secondary actuator design becomes a trade-off between jet pipe valve null flow, seal and linkage friction, and power valve shear-out force capability. The stepper motor approach is a compromise between speed of response (slew rate) and resolution.

Dynamic response of the EM approach is limited by the acceleration capability of the electric motor. The dominant inertia term in the EM design is the inertia of the electric motor. The load inertia when reflected through the actuator gearing (2000:1 typical) is negligible in comparison to the motor inertia. The speed of response is directly related to the square root of the motor torque-to-inertia ratio. Torque-to-inertia ratios of electric motors of the high power class fall in the 30 to 50 range. A typical value for a torque-to-inertia ratio for a hydraulic powered surface, where surface inertia completely dominates, is 500. The EM approach is response limited by the electric motor torque-to-inertia ratio whereas the hydraulic actuator approaches are not. The EM approach is rated fair.

3.4.2.6 Cost. Three aspects are associated with the cost of ownership of hardware: initial cost, maintenance cost, and cost incurred because of down time. Minimizing cost of ownership requires maximum utilization--the aircraft must be available continuously. Estimating production costs of undeveloped hardware is highly spectulative, however, among the hydraulic approaches the secondary actuator is expected to cost the most; hydraulic servcs are more expensive than torque motors and stepper motors. The PM torque motor approach is estimated to be the cheapest due to its simplicity and fewer parts. The EM approach appears competitive on an initial cost basis. The largest unknown is the motor controller.

Maintenance costs involve repair parts cost, maintenance manhours (maintainability), and the number of failures per flight hour (operational reliability). The simplest approach having the fewest most reliable parts is expected to have the lowest maintenance cost, therefore the PM torque motor approach is rated highest followed by the stepper motor and secondary actuator approaches, and lastly the EM approach primarily due to electric motor failure rates.

Down time cost involves major system elements (i.e., electronic package, actuator, etc.), system interconnect lines (electrical and hydraulic), and failure occurrence rate. All of the approaches lend themselves to fault isolation of major elements and then removal. Projected down time is therefore expected to be a function of interconnect lines and overall reliability. The PM torque motor approach has the advantage of requiring the fewest number of electrical lines, while the EM approach has the advantage of no hydraulic lines but must tolerate motor reliability.

Taking into account all the factors, the PM torque motor approach is rated excellent; the other two hydraulic approaches are rated good and the EM approach is rated fair.

3.4.2.7 Weight. Weight of the three hydraulic approaches are all fairly equal with the secondary actuator being the heaviest, and the stepper motor next. Considering that high pressure hydraulics offers potentially significant weight reduction, the secondary actuator becomes less competitive due to hardware and efficiency considerations. The EM actuation system is considerably heavier than the linear hydraulic actuation system for high power levels. EM does have some weight advantage in the transmission system area, but not enough to overcome a 2 to 1 weight disadvantage in the actuation area. (This does not include weight reductions made possible by utilizing high pressure hydraulics.)

The PM torque motor and stepper motor approaches are rated excellent, the secondary actuator approach is rated fair, and the EM approach is rated poor.

3.4.2.8 Limiting Factors. This term refers to the technology state of both the required components and integrated systems. Technology for the secondary actuator is rated excellent in both the component and systems area. The PM torque motor and stepper motor approaches are rated good; component technology is available but some system development is required. The EM approach is rated fair, primarily due to the uncertainties in the controller area. The development status would not preclude the use of any of the four concepts on the proposed Phase II flight application.

The comparative analysis is qualitative and therefore somewhat subjective, since all comparisions not based completely on physical facts (complete design, production and historical operational data) are subjective. Given sufficient time and effort a more detailed comparison could have been made, however, the end result is not expected to change significantly. The EM approach comparison is felt to be the most subjective simply because background information and operational experience regarding high response, high power, redundant EM systems is not available.

3.5 COMPARATIVE ANALYSIS RECOMMENDATIONS

Recommendations for future effort depend upon overall goals of the digital FBW program. The first task therefore is to establish realistic goals for actuation system technology commensurate with the overall program. A minimum effort approach to actuation is the secondary actuator concept; this requires no state-of-the-art advancement. By accepting additional tasks and effort, an advance in the state-of-the-art is offered by the direct drive actuator approach. With considerably more developmental effort and also risk, power-by-wire concepts hold promise of advancing the state-of-the-art and providing an alternate to hydraulic power systems. In relation to the current contracted effort, the following recommendations are enumerated:

a. Three design approaches -- two (2) direct drive and one (1) secondary actuator -- should be carried through in the conceptual design phase of this study. A quantitative comparison between the two direct drive approaches, PM torque motor and stepper motor, and the secondary actuator approach should be performed to select the best approach. The secondary actuator concept would be the baseline for comparison.

b. The electro-mechanical (EM) approach is recommended for further development. The effort envisioned is specific hardware design, development, and testing. This effort should be conducted and coordinated with the overall FBW program to produce flight-worthy hardware for incorporation into the F-8 test vehicle in the 1976 period. EM technology has progressed to the point where specific design and test data are required to advance the concept. The scope and extent of the envisioned effort required to produce a significant contribution is beyond that available on this contract, therefore, it is recommended that the EM approach be pursued under an extension of the present contract or under a separate contract. It is further recommended that this be a joint airframe/component manufacturer effort. This arrangement has the advantage of combining the expertise of both the airframe manufacturers and the component manufacturers in establishing a total integrated system design properly accounting for primary control design and verification, power generation and distribution, flying qualities, duty cycle, structural design, system design, and component design.

c. A study of the entire power generation and distribution systems in aircraft should be conducted. The study should consider: weight, efficiency, reliability, maintainability, single and multi-engine aircraft, high voltage AC and DC high frequency power, centralized and localized power distribution, very high pressure hydraulics, and integrated actuator packages. The integrated actuator package offers the potential of high level of reliability by sealing the package against outside contamination and eliminating the human element in maintenance. This concept combines many desirable features:

1. The advantages of electrical power distribution system.

2. Compatibility with high voltage power systems.

3. The high performance features of hydraulics.

4. Elimination of contamination problems.

5. Increased overall reliability and simplified field maintenance.

6. Elimination of servo amplifier and necessary power supplies, etc., since control signal power is derived directly from the computer power supply.

7. Overall system weight reduction (touted by IAP proponents).

Advancement of the state-of-the-art in this area is beyond the scope of the present contract, therefore, it is recommended that pursuit of this approach be done separately.

SECTION 4

CONCEPTUAL DESIGN

4.1 INTRODUCTION

Three of the most promising actuation design concepts were selected with the concurrence of NASA for more detailed study and preliminary design. The three system concepts were:

(1) a PM Torque Motor Direct Drive System

(2) a Stepper Motor Direct Drive System

(3) an Improved Secondary Actuator System

The secondary actuator approach was carried through the conceptual design phase primarily as a baseline for comparison. New secondary actuators (other than the present NASA-F8-FBW aircraft units) were assumed for purpose of discussions. The results of the more detailed study and preliminary design effort are reported in this section of the report. This effort consisted of:

- Preliminary system trade-off study to consider factors involved in application of the design concepts to the pitch, roll, and yaw control of the Phase II flight aircraft. This trade-off considered use of existing aircraft power control hydraulic systems, redundancy provisions to assure flight control actuation system reliability, performance, economic factors, and interface requirements.

- Preliminary design of the digitally controlled actuation system for primary flight control functions for the Phase II aircraft. Packaging, sizing, and installation factors were considerations with the intent to identify problem areas which would preclude development of the concept for use in the Phase II aircraft.

- Accumulation of comparative data on the alternate design concepts to permit selection of the most promising design concept for subsequent design and development. The PM torque motor direct drive approach appears to best satisfy the requirements for the proposed application and is recommended for further development.

  Development of mathematical models of the PM torque motor direct drive actuation system for real-time computer simulation on a hybrid computer.

The effort reported upon in Section 4 was accomplished after that reported upon in Section 3, and after an interim briefing; consequently there is some overlap in reporting. The relation of this effort to the total program effort is depicted in Figure 4.1-1.

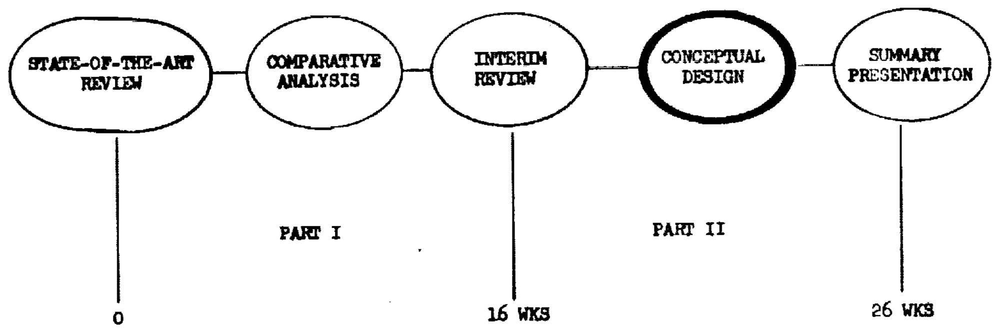


FIGURE 4.1-1 DIGITAL FLIGHT CONTROL ACTUATION SYSTEM PROGRAM

4.2 F-8 APPLICATION TRADE-OFF STUDY

The three designs selected in Section 3 were developed in greater detail for application in the F-8 aircraft for the specific purpose of selecting the most appropriate actuation system for use in a multichannel digital FBW system. The factors considered in this trade-off were: performance, redundancy and reliability, system interface, power systems impacts, and economic factors. These factors are discussed in detail in the following paragraphs. The trade study indicated the PM torque motor offers considerable advantage over both the secondary actuator and stepper motor approaches in performance, reliability, and simplicity. It is compatible with either the three or four channel systems. The actuation system can be developed to meet the requirements of the digital fly-by-wire program.

4.2.1 PERFORMANCE

An improved secondary actuator system composed of an electro-hydraulic servo positioning a spool/sleeve valve is the first system considered. A block diagram representation of this type system is depicted in Figure 4.2-1.

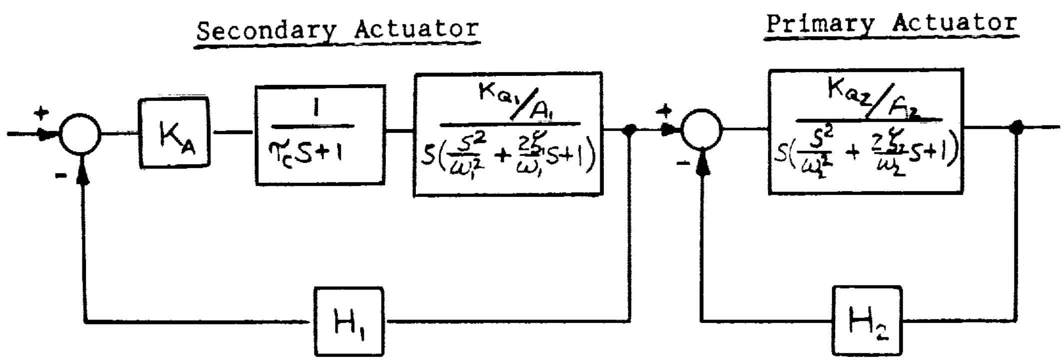
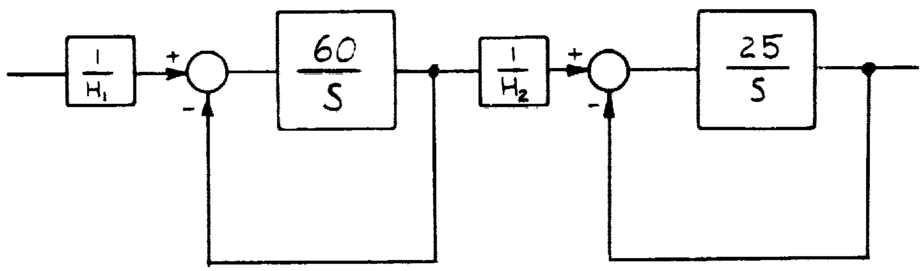


FIGURE 4.2-1 SECONDARY ACTUATOR APPROACH - BLOCK DIAGRAM

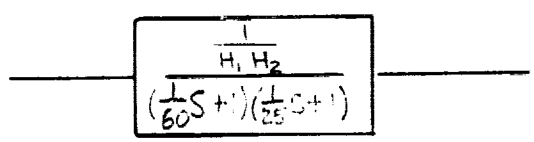
The valve has a mechanical natural frequency (ω_1) in the neighborhood of 100 to 200 hertz and is well damped (ζ = .7 to .9). The coil has a time constant in the neighborhood of 4 msec. The mechanical resonance is far above the passband and can be ignored; the electrical time constant being much nearer must be taken into account. A conventional approach to overcoming the electrical time constant is to drive the valve with a current amplifier. With sufficient supply voltage, a current amplifier can essentially eliminate the electrical time constant. Loop gains for secondary actuators generally range from 40 to 80 sec^{-1}.

The actuator natural frequency (ω_2) is a function of the effective hydraulic spring rate, the structural stiffness, and surface inertia. The natural frequencies are generally above 30 hertz and can be ignored for a first order approximation.

The $H \frac{K_Q}{A}$ terms are the respective loop gains. Typical loop gains for primary actuators range from 20 to 30 sec^{-1}. Assuming values of 25 for the primary loop and 60 for the secondary loop produces:



which reduces to:



Frequency response of this representation is shown in Figure 4.2-2. It can be seen that ideally the system would just meet the .52 rad (30°) phase shift requirement at 8.9 rad/s and thus would not meet the baseline bandwidth requirement of 13.5 rad/s established for the horizontal axis.

The secondary actuator loop gain could perhaps be increased somewhat to improve the performance. With a loop gain of 80, the .52 rad (30°) phase lag point would occur at 9.5 rad/s. The phase contribution of the secondary actuator can be seen to decrease rather slowly as the secondary actuator pole moves out in frequency. This simple analysis ignores a number of characteristics which become increasingly important as the loop gain increases and as the actuator size decreases; namely, backlash and friction. The nonlinearities have the effect of introducting low frequency phase shift and reducing loop gain. For instance, "O" ring seal friction in actuators can easily introduce .2 to .89 kN (50 to 200 lbs) friction. This would be intolerable in an actuator designed for 1.8 kN (400 lbs) maximum force output. Special seals would be necessary to obtain reasonable performance.

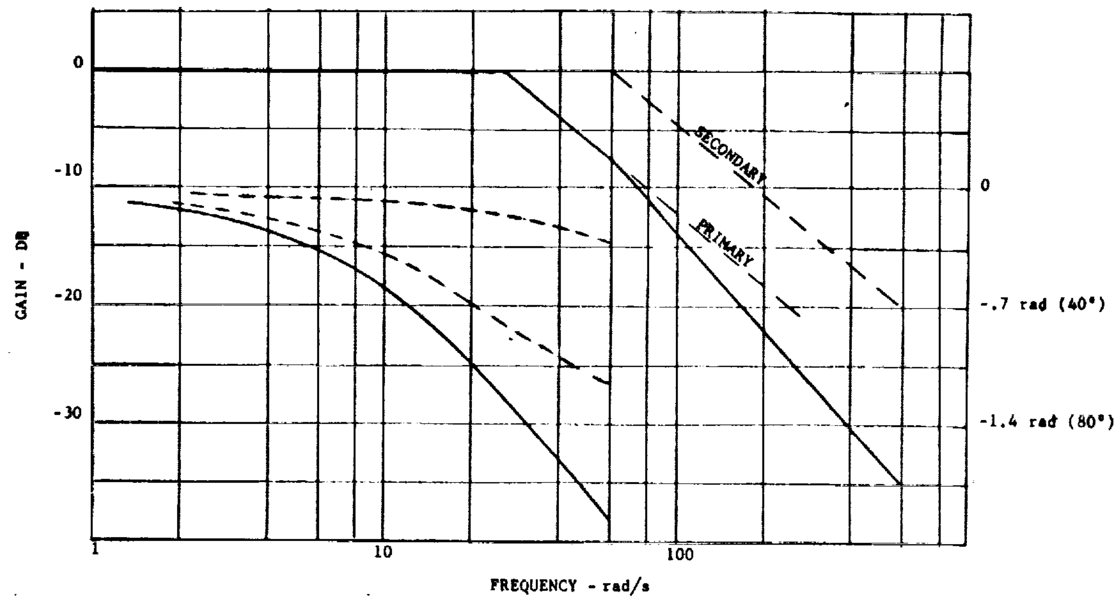


FIGURE 4.2-2 SECONDARY ACTUATOR RESPONSE

The problem of friction in small electro-hydraulic servo actuators can be seen by considering the model shown in Figure 4.2-3. The nonlinearity, stiction, and coulomb friction has been combined in a single element. The stiction and coulomb terms X_s and X_c, as defined in Figure 4.2-3, represent friction forces scaled by the appropriate servo gains. Gain and phase of the describing function for this nonlinearity is plotted in Figure 4.2-4 as a function of the ratio $\frac{X_c}{\varepsilon}$. The value of the describing function is a function of both the

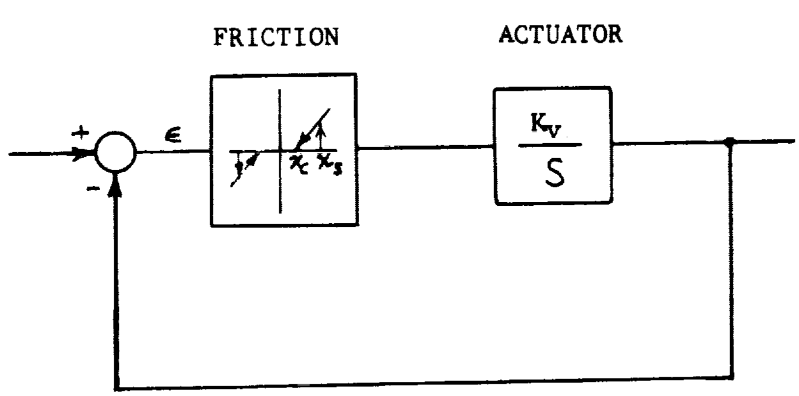


FIGURE 4.2-3 SERVO ACTUATOR WITH FRICTION

input signal amplitude and the ratio of the striction and coulomb friction forces. For purposes of discussion, consider the following values for small secondary actuators:

$$K_V = 122$$

$$\frac{X_s}{X_c} = 2$$

$$X_s = F_s/K_A\ K_p\ A_p$$

$$= .002 \text{ in. (corresponding to 16 lbs.)}$$

Stroke = 50.8 mm (2 in.)

Where:

K_V is the velocity constant

$\frac{X_s}{X_c}$ is the ratio of stiction to coulomb friction

F_s is the stiction force

K_A is the amplifier gain

K_p is the servo valve pressure gain

A_p is the actuator piston area

Closed loop response of this system with and without the nonlinear friction term for 0.5% stroke input amplitude is shown in Figure 4.2-5. The nonlinear element gain and phase in effect vary with frequency and amplitude due to the system error. Input to the nonlinearity increases as a function of frequency, resulting in increased gain and reduced phase for the nonlinear element. The major effect on response is in low frequency phase shift. A secondary actuator approach using this actuator could not meet the .52 rad (30°) phase shift requirement at any frequency.

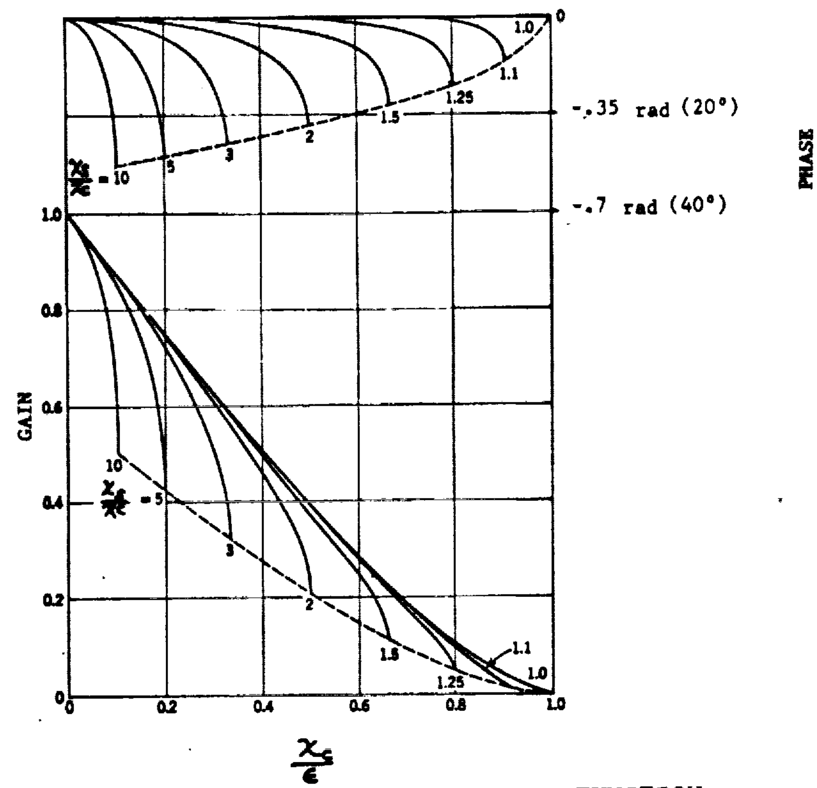


FIGURE 4.2-4 FRICTION DESCRIBING FUNCTION

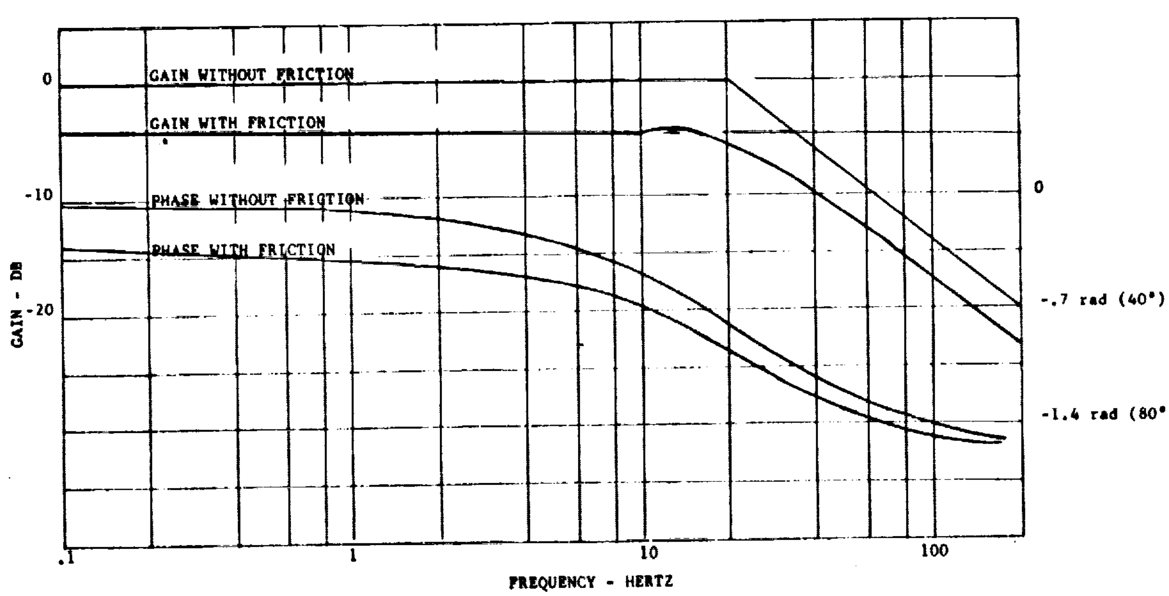


FIGURE 4.2-5 SERVO ACTUATOR WITH FRICTION RESPONSE

System response of the secondary actuator can be improved by increasing the primary actuator loop gain. This requires either an increase in the effective hydraulic natural frequency or damping, which can only be accomplished with increased weight and/or power penalties. Natural frequency is ultimately limited by structural stiffness. Optimum design dictates minimum weight and maximum efficiency. Increasing system response by increasing the primary actuator loop response is an expensive approach.

A better and easier method of obtaining high performance is to utilize a high response direct drive valve eliminating the secondary actuator entirely. The PM torque motor direct drive valve is such a device. Mechanical resonance for large torque motors fall in the 200 to 500 hertz range, far beyond the passband. A block diagram representation of the PM torque motor direct drive actuation system is illustrated in Figure 4.2-6.

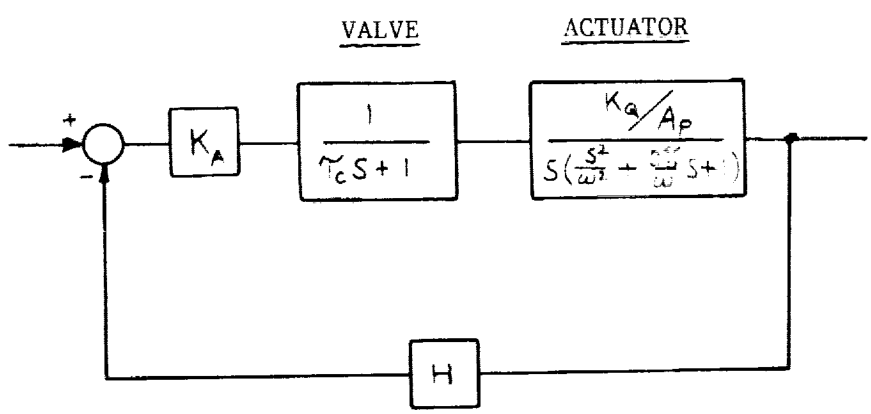


FIGURE 4.2-6 PM TORQUE MOTOR APPROACH - BLOCK DIAGRAM

The electrical time constant is about 9 ms and can be circumvented by utilizing a current amplifier. Again, neglecting the high frequency resonance and assuming a primary loop gain of 25, the frequency response is depicted in Figure 4.2-7. The .52 rad (30°) phase shift point occurs beyond 15 rad/s. Increasing loop gain to 30 will move the .52 rad (30°) phase shift point out to 18 rad/s. The single control loop permits very high performance, limited ultimately by structural and power considerations.

The PM torque motor direct drive approach provides considerable improvement in dynamic response while reducing weight and power dissipation. The PM torque motor direct drive approach can easily meet the high performance requirements specified in Section 3.

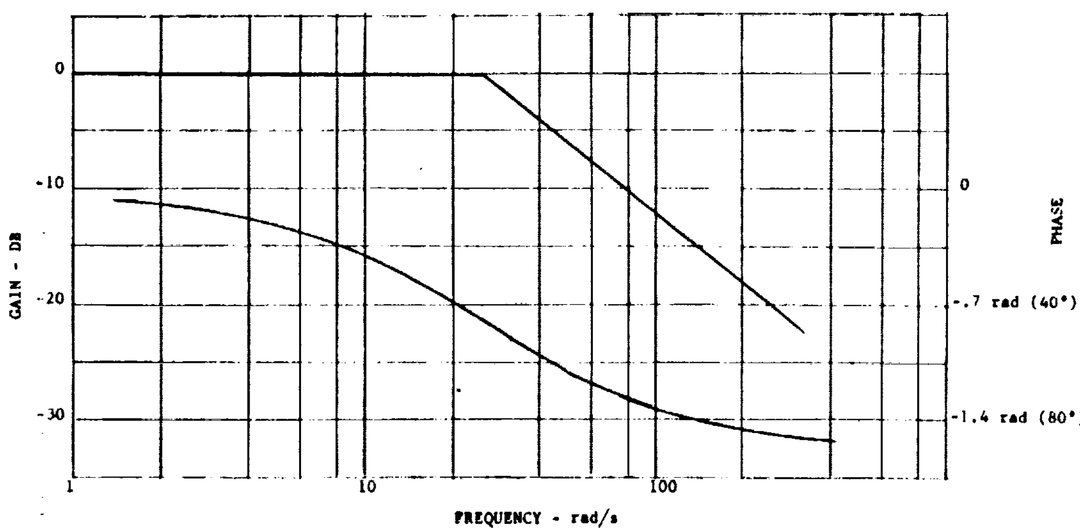


FIGURE 4.2-7 PM TORQUE MOTOR APPROACH RESPONSE

Response is a critical factor in utilizing the stepper motor in a direct drive valve application. Specific design of the controller and the load are both quite significant in attaining high response. In general, variable reluctance stepper motors, as compared with permanent magnet stepper motors, have lower rotor inertias, higher speed capability and are simpler in construction. They are lower in efficiency, have lower damping, and are not presently available in as high a power units.

Dynamic response of a stepper motor to a single step input is characterized by a poorly damped quadratic having a natural frequency in the neighborhood of 150 hertz. For multiple steps, succeeding pulses cannot be commanded until the motor has moved sufficiently, thus there is a maximum pulse rate which the motor can follow without loosing synchronism. For multiple step inputs, the pulse rate, and consequently motor speed, must be limited to a rate referred to as the start-stop mode rate. The motor can accept a higher pulse rate as the rotor speed increases. The maximum rate capability of the stepper motor is referred to as the slew rate.

Stepper motors can operate at a higher speed by increasing the pulse rate gradually. One method of doing this is to filter the pulse rate command signal for high rate inputs. The minimum filter time constant for the selected motor is 25 ms . Preliminary design effort, using a very high response motor, 2000 pps start-stop rate and an 8000 pps slew rate, leads to a configuration whose dynamic model is depicted in Figure 4.2-8.

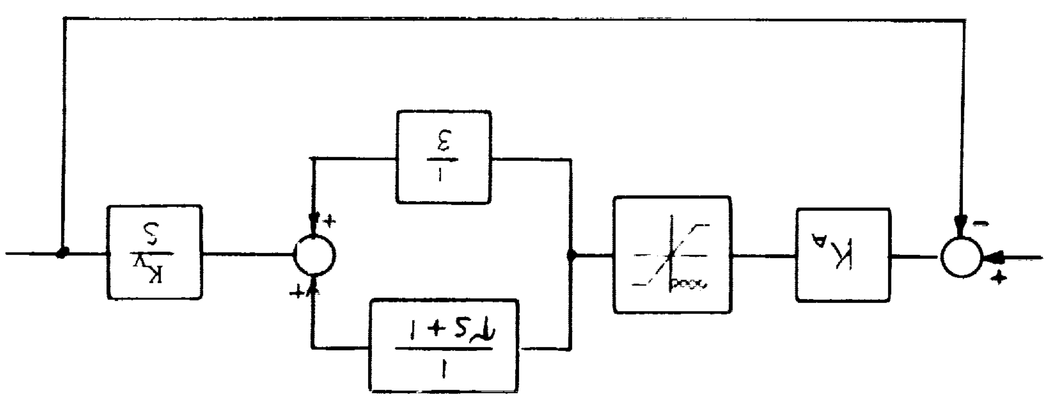


FIGURE 4.2-8 STEPPER MOTOR DYNAMIC MODEL

Two features were incorporated to increase the response: first, ramping control was incorporated to obtain higher rates, and second, a two stage servo valve was incorporated. The two stage valve permits a higher gear ratio since the shear-out force requirement does not dictate gearing. Higher slew rates are possible. A loop gain was established to provide high band pass. It is estimated that this approach would be capable of response equivalent to that of a double pole at a corner frequency of 30 hertz. The actuation system model, assuming a primary actuator loop gain of 25 is shown in Figure 4.2-9 and its frequency response is shown in Figure 4.2-10. From Figure 4.2-10, it can be seen that the .52 rad ($30°$) phase shift point occurs at 9.4 rad/s.

The foregoing analysis shows that the PM torque motor approach is inherently capable of the highest response, being limited only by the power actuator. The secondary actuation system approach is capable of meeting the basic F-8 response requirements, but cannot meet the study baseline requirements set for future aircraft employing advanced control laws. The conventional secondary actuator is inhereintly less responsive than the PM torque motor approach. The stepper motor can provide response comparable to the conventional secondary actuator approach.

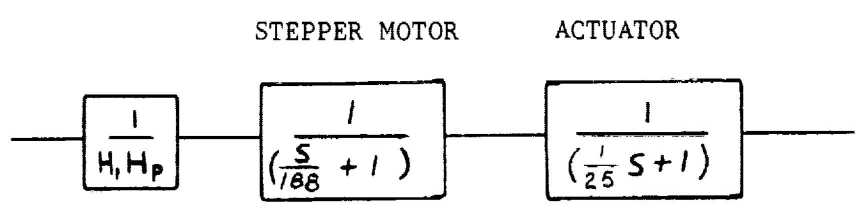


FIGURE 4.2-9 STEPPER MOTOR APPROACH MODEL

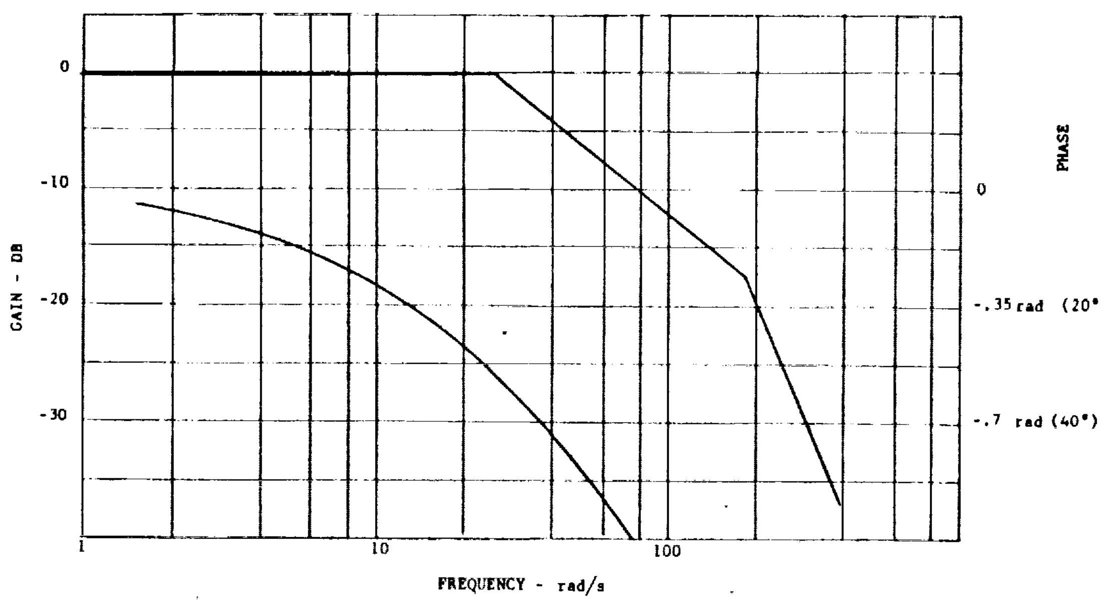


FIGURE 4.2-10 STEPPER MOTOR APPROACH RESPONSE

4.2.2 REDUNDANCY AND RELIABILITY

Reliability diagrams of the secondary actuator approach are shown in Figure 4.2-11 and Figure 4.2-12. Considering only one failure state (open), then the operational reliabilities for the three and four channel systems can be shown (Appendix G) equal to:

$$R \approx 1 - Q_B - 3Q_A^2 \quad @ \quad 3 \text{ Channels}$$

$$R \approx 1 - Q_B - 4Q_A^3 \quad @ \quad 4 \text{ Channels}$$

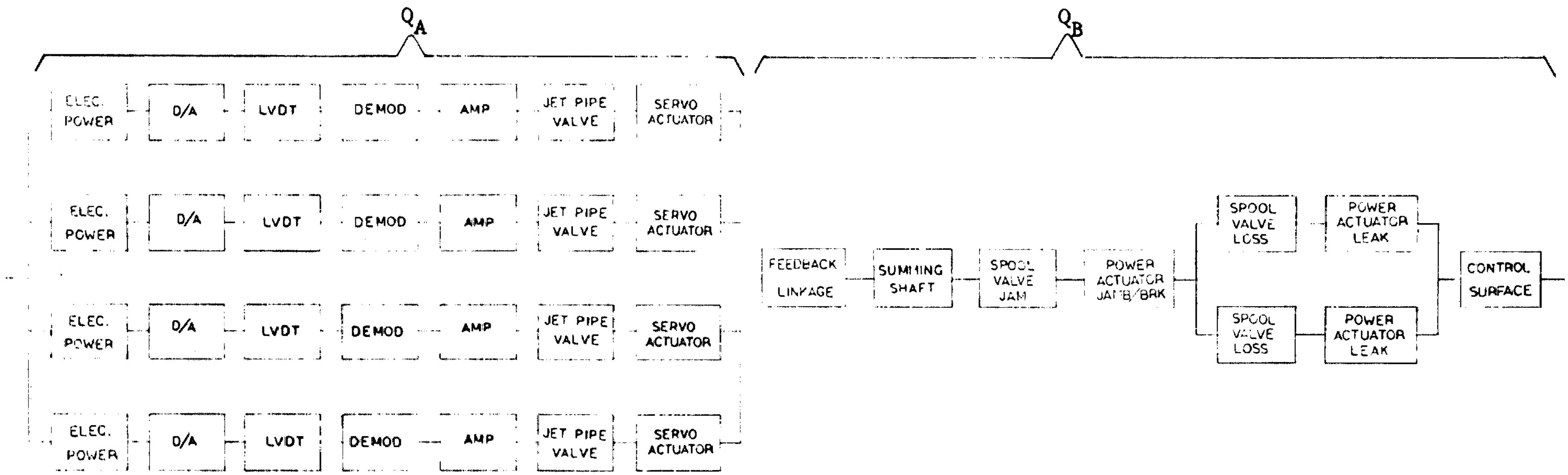


FIGURE 4.2-11 SECONDARY ACTUATOR APPROACH RELIABILITY DIAGRAM

Operational reliability is defined as the probability that the system will be operational at the end of a one hour period; the operational state being defined as when some level of control is afforded. Q_A and Q_B are the probabilities that the corresponding elements will fail within the specified time period. It can be seen that the four channel system is slightly more reliable. The reliability of both are predominantly determine by the series elements (B); namely, the actuator and hydraulic supplies. Channel failure contributions are relatively small and insignificant in comparison.

A more representative view of hardware reliability for FBW purposes is to consider failures to fall into three categories: opens, negative hardovers, and positive hardovers. It can be shown, by analyzing system reliability in terms of four states, the operation state and the three failure states, and defining the system failure event as being when total loss of control occurs, that the overall operational reliability is equal to:

$$R \approx 1 - Q_B - 2Q_A^2 \quad @ \quad 3 \text{ Channels}$$

$$R \approx 1 - Q_B - \frac{4}{3} Q_A^2 \quad @ \quad 4 \text{ Chanels}$$

This is based upon an equally probable failure distribution; that is, an element is as likely to fail open as it is to fail hardover in the positive direction as it is to fail hardover in the negative direction. It is interesting to note that the three channel system is now nearly as reliable as the four channel system. Both are again primarily dependent upon the series elements (B), the actuation and hydraulic supplies.

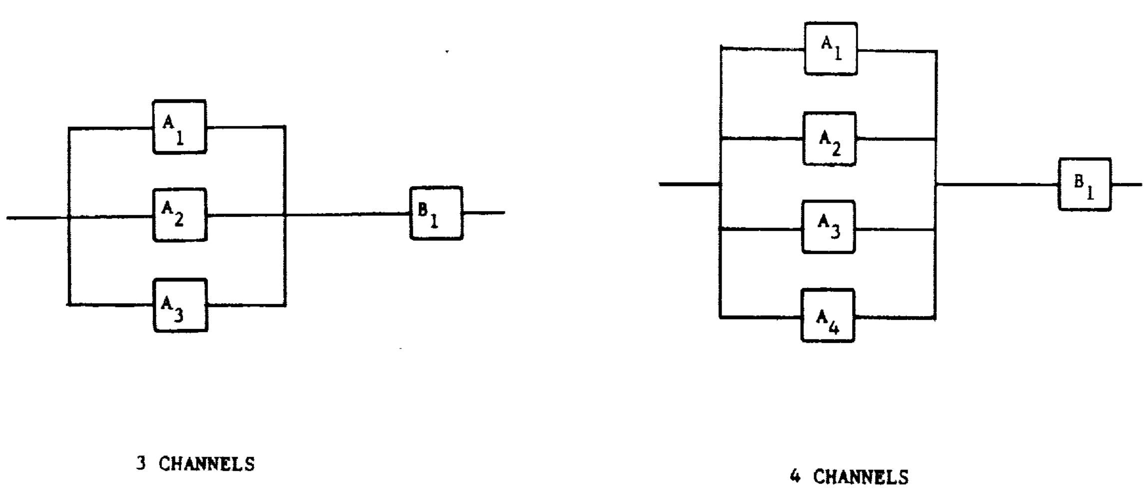


FIGURE 4.2-12 SECONDARY ACTUATOR APPROACH - SIMPLIFIED RELIABILITY DIAGRAM

A comparison of three and four channel secondary actuation systems which incorporate channel comparison monitors is given below. The monitor determines and disconnects failed channels, by majority vote, until only two channels remain connected; it then ceases to monitor off additional channels.

$$R \approx 1 - Q_B - 2\,Q_A^2 \quad @ \quad 3 \text{ Channels}$$

$$R \approx 1 - Q_B - \frac{8}{3}\,Q_A^3 \quad @ \quad 4 \text{ Channels}$$

Note that the operational reliability of the three channel system is not improved by the monitor. An improvement is indicated in the four channel system, however, considering only the actuation system it is still insignificant since the series elements (B) predominates. In addition, the monitor, having disconnect authority, introduces additional elements and although small, probably cancels the small improvement shown.

The reliability diagram for the PM torque motor direct drive approach is shown in Figure 4.2-13 and Figure 4.2-14. In comparing these diagrams with the secondary actuator diagrams, it can be seen that the significant difference is in the individual elements. When simplified by combining the parallel 'B' elements, it reduces to a form identical to the secondary actuator approach (i.e. Figure 4.2-12); that is, three or four parallel elements in series with a series element. Redundant channel amplifiers, power supplies, and coils essentially eliminate their influence in the overall reliability. Channel failure rate is therefore primarily a function of the LVDT's, demodulators, and D/A converters. Comparison of the individual element reliabilities is presented in Table 4.2-1.

TABLE 4.2-1 FAILURES RATES - SECONDARY ACTUATOR AND PM TORQUE MOTOR

	SECONDARY ACTUATOR	FAILURES PER 10^6 HRS	PM TORQUE MOTOR	FAILURES PER 10^6 HRS
'A'	Electrical Power and Amplifier*	.002	Electrical Power, Amp. and Torque Motor Coil*	0.0025
	D/A Converter	3.0	D/A Converter	3.0
	J.P. Valve	50.0		
	Servo Actuator	1.0		
	LVDT Sensor	3.0	LVDT Sensor	3.0
	Demodulator	4.0	Demodulator	4.0
	TOTAL	71.0	TOTAL	10.0
'B'	Fdbk Linkage	.01	Torque Motor	.01
	Summing Shaft	.01	Valve Linkage*	
	Spool Valve Jam	.2	Spool Valve Jam	.2
	Power Act. Jam/Break	.35	Power Act. Jam/Break	.35
	Spool Loss & Act. Leak*	.001	Spool Loss & Act. Leak*	.001
	Control Surface	.01	Control Surface	.01
	TOTAL	.57	TOTAL	.56

*Dual Redundant Equivalent

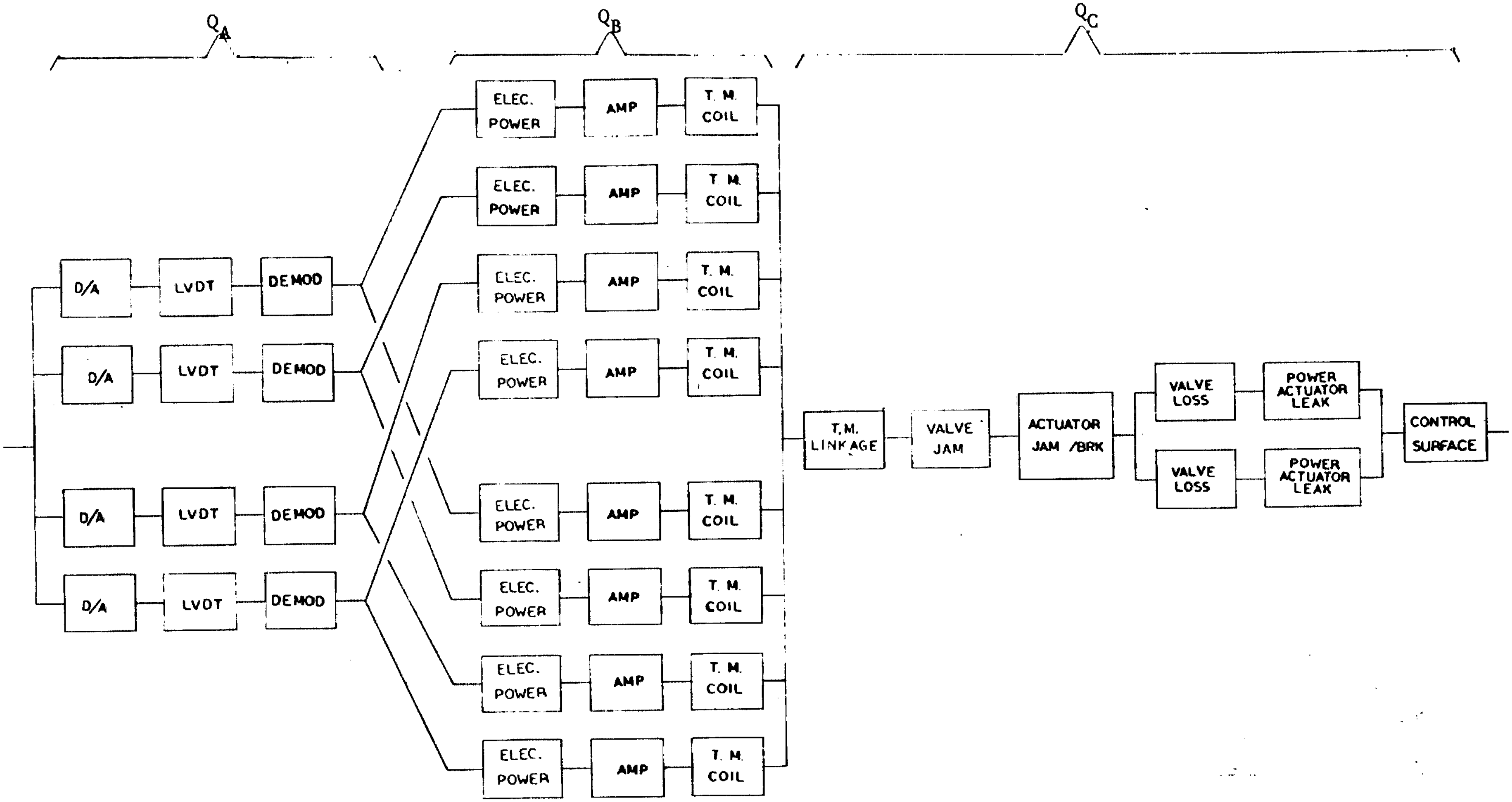
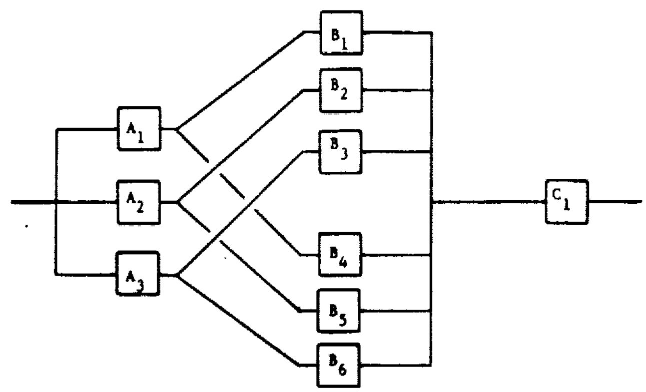
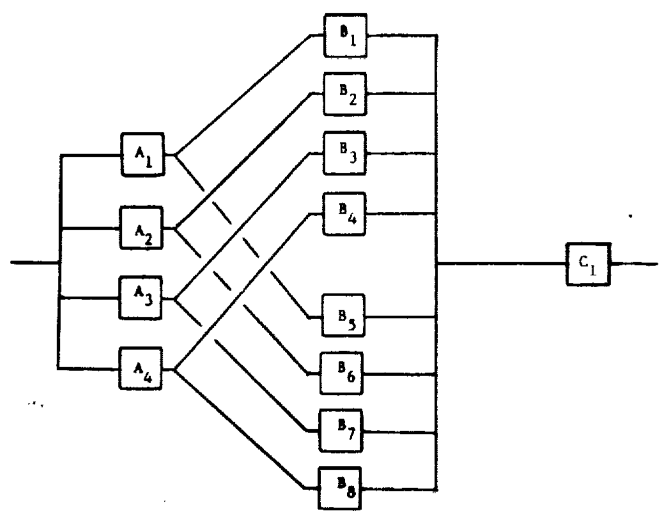


FIGURE 4.2-13 PM TORQUE MOTOR DIRECT DRIVE CONCEPT



3 CHANNEL



4 CHANNEL

FIGURE 4.2-14 PM TORQUE MOTOR APPROACH - SIMPLIFIED RELIABILITY DIAGRAM

The preceding analysis shows the power actuators are series elements in the reliability diagram and that they essentially establish the overall actuation system operational reliability.

Past history of power actuation components (Appendix E and Bibliography 043) show that the major power actuator failure mode is a structural break, predominately rod ends. The average catastrophic failure rate, for Naval fleet operations, is 2.78×10^{-6} FPFH. It is obvious the actuation system reliability goal specified in Section 3 cannot be met with average tandem actuators. Reliability of the tandem hydraulic actuator can be improved by a number of techniques; such as, rip-stop design, increased structural margins, improvement or elimination of conventional rod ends, steel actuator bodies and thorough parts inspection, to name a few. It is reasonably estimated that the actuator structural failure rate can be reduced considerably below the fleet average to 0.35×10^{-6} FPFH. Consequently a value of 0.35×10^{-6} FPFH is estimated and used in the preceding calculations.

The stepper motor approach reliability diagram is shown in Figures 4.2-15 and 4.2-16. The stepper motor approach is dependent upon the monitor which is also redundant. The monitor in conjunction with the inherent characteristics of the stepper motor essentially eliminates hardover potential. The imperfection in the monitor is assumed to be adequately represented by the series element labled MONITOR. Table 4.2-2 summarizes the failure rates. Considering the simplified reliability diagram in Figure 4.2-16, it can be shown (Appendix G) that the probability of having some level of control at the end of one hour is:

$$R \approx 1 - Q_C - Q_B^2$$

which again depends primarily upon the series element (C).

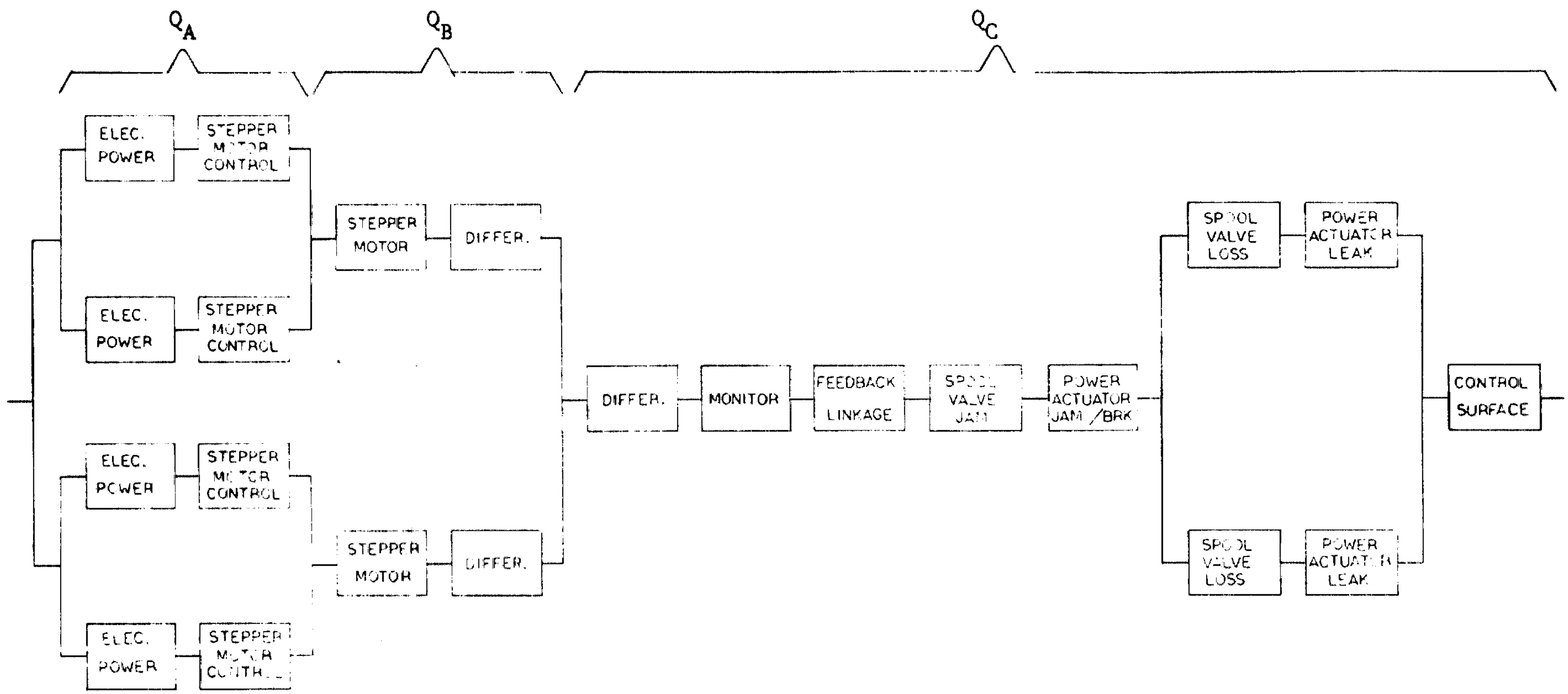


FIGURE 4.2-15 DIRECT DRIVE STEPPER MOTOR

TABLE 4.2-2 OPERATIONAL FAILURE RATE - STEPPER MOTOR APPROACH

		FAILURES PER 10^6 HRS
A	Electrical Power	35
	Stepper Motor Control	79
	TOTAL	114
B	Stepper Motor	25.0
	Differential	.06
	TOTAL	25.06
C	Differential	.03
	Monitor (Est.)	.01
	Feedback Linkage	.02
	Spool Valve Jam	.2
	Power Act. Jam/Break	.35
	Spool Valve Loss & Act. Leak*	.001
	Control Surface	.01
	TOTAL	0.62

*Dual Redundant Equivalent

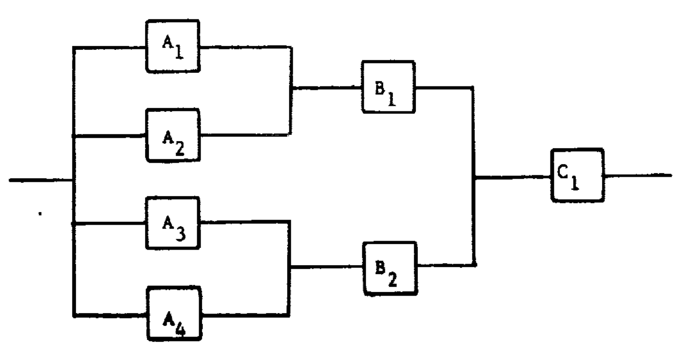


FIGURE 4.2-16 STEPPER MOTOR APPROACH - SIMPLIFIED RELIABILITY DIAGRAM

Comparing the three actuation system approaches, Table 4.2-3, it can be seen the PM torque motor approach has the highest reliability followed by the secondary actuator and lastly by the stepper motor approach. The differences are minor and all three approaches are essentially dependent upon the series elements, primarily the hydraulic actuator.

TABLE 4.2-3 COMPARISON OF ACTUATION SYSTEM OPERATIONAL FAILURE RATES

ACTUATION SYSTEM	OPERATIONAL FAILURE RATE
PM Torque Motor	0.56×10^{-6} FPFH
Secondary Actuator	0.57×10^{-6} FPFH
Stepper Motor Approach	0.62×10^{-6} FPFH

The maintenance reliability is directly related to the system MTBF. The total parts failure rates for the three approaches are summarized in Table 4.2-4.

TABLE 4.2-4 COMPARISON OF ACTUATION SYSTEM MAINTENANCE FAILURE RATES

ACTUATION SYSTEM	3 CHANNELS	4 CHANNELS
PM Torque Motor Direct Drive	860×10^{-6}	930×10^{-6} FPFH
Secondary Act.	1090×10^{-6}	1240×10^{-6} FPFH
Stepper Motor Direct Drive	1010×10^{-6}	1120×10^{-6} FPFH

The conclusions to be drawn from the foregoing reliability analysis are:

1. Operational or failsafe reliability of the actuation system alone is essentially equal to the reliability of the power actuator. The difference between 3 and 4 channel control is relatively insignificant.

2. The PM torque motor direct drive approach will have significantly higher MTBF than either the secondary actuator or the stepper motor approaches.

3. Meeting the reliability goal specified in Section 3 will require considerable improvement in hydraulic power actuators. Proper design, suitable margins, and adequate inspection and testing will insure that the actuator meets these requirements.

If the entire FBW system is considered, then the major elements which must be added are the computers and IMU's, the electrical and hydraulic power generation and distribution systems and mechanical input controls (i.e., control stick, etc.). Projected failure rates for these items are as follows: the computers and IMU's, a combined overall failure rate of 1000×10^{-6} FPFH per channel; the mechanical input controls, a failure rate of 0.01×10^{-6} FPFH; and electrical system, a failure rate of 127×10^{-6} FPFH. The reliability diagram of the hydraulic power systems for the F-8C is shown in Figure 4.2-17.

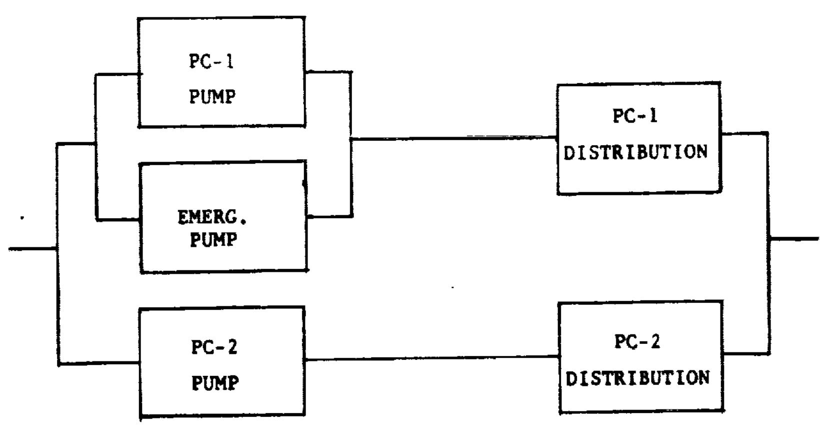


FIGURE 4.2-17 F-8C HYDRAULIC POWER SYSTEM - RELIABILITY DIAGRAM

The probability of having hydraulic power for a one hour period is:

$$R \approx 1 - Q_P Q_D - Q_D^2$$

where Q_P is the probability of a pump failing within the one hour period and Q_D is the probability of a distribution system failing in the one hour period. Using the reliability numbers as discussed in paragraph 3.3.1.1.4 produces:

$$R \approx 1 - 1.38 \times 10^{-6}$$

The total FBW system reliability diagram, shown in Figure 4.2-18, is based upon total computer, IMU and input sensor monitoring which provides control as long as one computer is still in operation, upon three primary AC power systems (or two plus batteries), and upon the F-8C type hydraulic power system. Using the individual probability numbers as previously delineated the system operational reliability is given by:

$$Q \approx Q_{STICK} + Q_{SENS} + Q_{ELEC} + Q_{COMP} + Q_{ACT} + Q_{SURFACE}$$

$$\approx (.01 + .03 + .002 + .001 + .56 + .01) \times 10^{-6}$$

$$\approx 0.613 \times 10^{-6}$$

The system operational reliability, using the PM torque motor direct drive valve approach, exceeds the 0.67×10^{-6} FPFH goal specified in Section 3.0. The entire system including the hydraulic system, has a failure probability of 1.993×10^{-6} FPFH. Comparable figures for the secondary actuator and stepper motor approaches are: 0.623×10^{-6} and 2.003×10^{-6}, and 0.673×10^{-6} and 2.053×10^{-6} respectively.

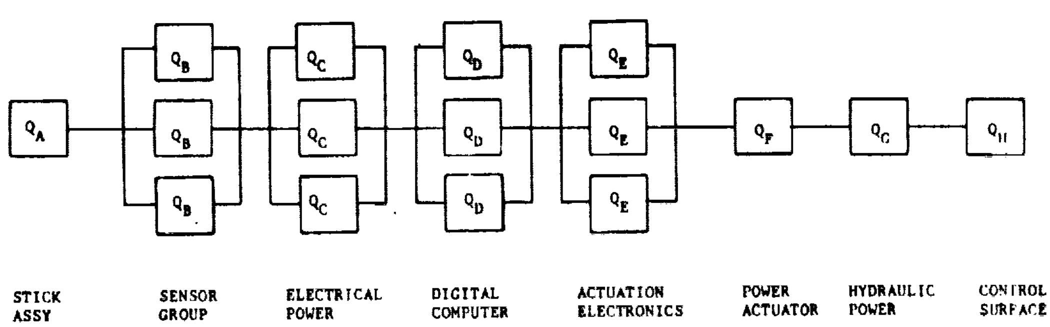


FIGURE 4.2-18 FLIGHT CONTROL SYSTEM RELIABILITY DIAGRAM

4.2.3 INTERFACE

A detailed study of the interface between the Digital Computer system and the actuation system requires design of the entire system. This investigation was therefore limited to a general study from the actuation system point of view. The following factors were considered to determine what if any advantage was offered by the three approaches in this area: number of channels, the valve drivers, interface electronics, loop closure, and monitor. The following paragraphs discuss these factors for each of the three approaches.

4.2.3.1 Number of Channels. All three approaches considered depend upon a single dual tandem power actuator. As such all are dependent upon a single mechanical channel. The number of channels referred to herein is the number of electronic channels controlling the dual tandem actuator. The primary criteria for selecting the number of channels are reliability, both operational and maintenance, and compatibility with the flight control computer system. Some flight control system designs dictate the number of channels to be carried through the actuation system,others do not.

The reliability and redundancy aspects of each approach have been covered in section 4.2.2. Restating, actuation system reliability is primarily a function of the power actuator and valve. Dependency of the three approaches to the number of channels is reflected in the contribution made by the electronic and valve driver channels to the total probability of an operational failure as shown below.

APPROACH	3 CHANNELS	4 CHANNELS	DIFFERENCE
Secondary Actuator	.5%	.34%	.16%
PM Torque Motor	.035%	.024%	.011%
Stepper Motor	.47%	.10%	.37%

It can be seen that they contribute little to the system operational failure probability. The difference between 3 and 4 channels is even less significant. The stepper motor approach does not lend itself to as logical a division by three as the other approaches.

The number of channels has a very significant effect upon the maintenance reliability figure as shown below.

APPROACH	3 CHANNELS	4 CHANNELS	DIFFERENCE
Secondary Actuator	1090×10^{-6}	1240×10^{-6}	12%
PM Torque Motor	860×10^{-6}	930×10^{-6}	7.5%
Stepper Motor	1010×10^{-6}	1120×10^{-6}	10%

The difference between 3 and 4 channels is roughly 10%. From an actuation system reliability point of view, three channels is preferred. A fourth channel increases the maintenance failure rate while having a minimal effect upon operational reliability. The optimum number of channels to be carried through the actuation system is dictated by considerations other than simply by the actuation system.

4.2.3.2 Valve Driver. In the secondary actuator approach the power control valve is driven by the force summed output of three or four secondary actuator elements. Two typical examples of this approach are the secondaries used in the NASA-F-8C and Air Force 680J FBW programs. These units are depicted in Figure 4.2-19 and 4.2-20. Physical data are tabulated in Table 4.2-5 and frequency response data are shown in Figure 4.2-21. These actuators are described in References (032),(142), (131), and (142). The problem in achieving high frequency performance with secondary actuators is illustrated in these two designs. If .17 rad (10°), of the .52 rad (30°) phase shift specified for the actuation system were allocated to the secondary actuator, then the 680J actuator would have a 2.5 hertz bandwidth, and the present F-8C actuators would be unable to provide the desired performance.

Mechanical aspects of the dual PM torque motor valve drive assembly are shown in Figure 4.2-22. The PM torque motor is functionally illustrated in Figure 4.2-23. The permanent magnet torque motor is the most commonly used electromechanical transducer for servo valve drives. These devices produce rotary motion which is translated into limited travel linear motion by a moment arm. The conventional torque motor utilizes a bridge magnetic circuit in which the armature tips move parallel to the air gaps flux. An advantage of the bridge circuit is that armature flux always passes through the permanent magnet in a re-enforcing direction, and therefore does not tend to demagnetize the unit. Typically, the armature is restrained by a spring within the air gap. As current flows through the armature winding, torque is produced which acts against the spring and produces displacement. An approximately linear angular displacement vs. current is produced.

Development of high torque requires high flux densities, and long strokes require large air gaps -- somewhat opposing requirements, since flux varies inversely with air gap distance. Usefulness of a permanent magnet is measured by the amount of flux it can produce in an air gap and the magnetic intensity it can maintain across the gap. Twice the maximum gap energy per cubic inch of magnet material is a figure-of merit known as "energy product". Cobalt samarium has an energy product in the .12 to .16 MT•A/m (15 to 20 million gauss-oersteds) range as compared with .03 to .04 MT•A/m (4-5 million gauss-oersteds) for other magnet materials (Alnico

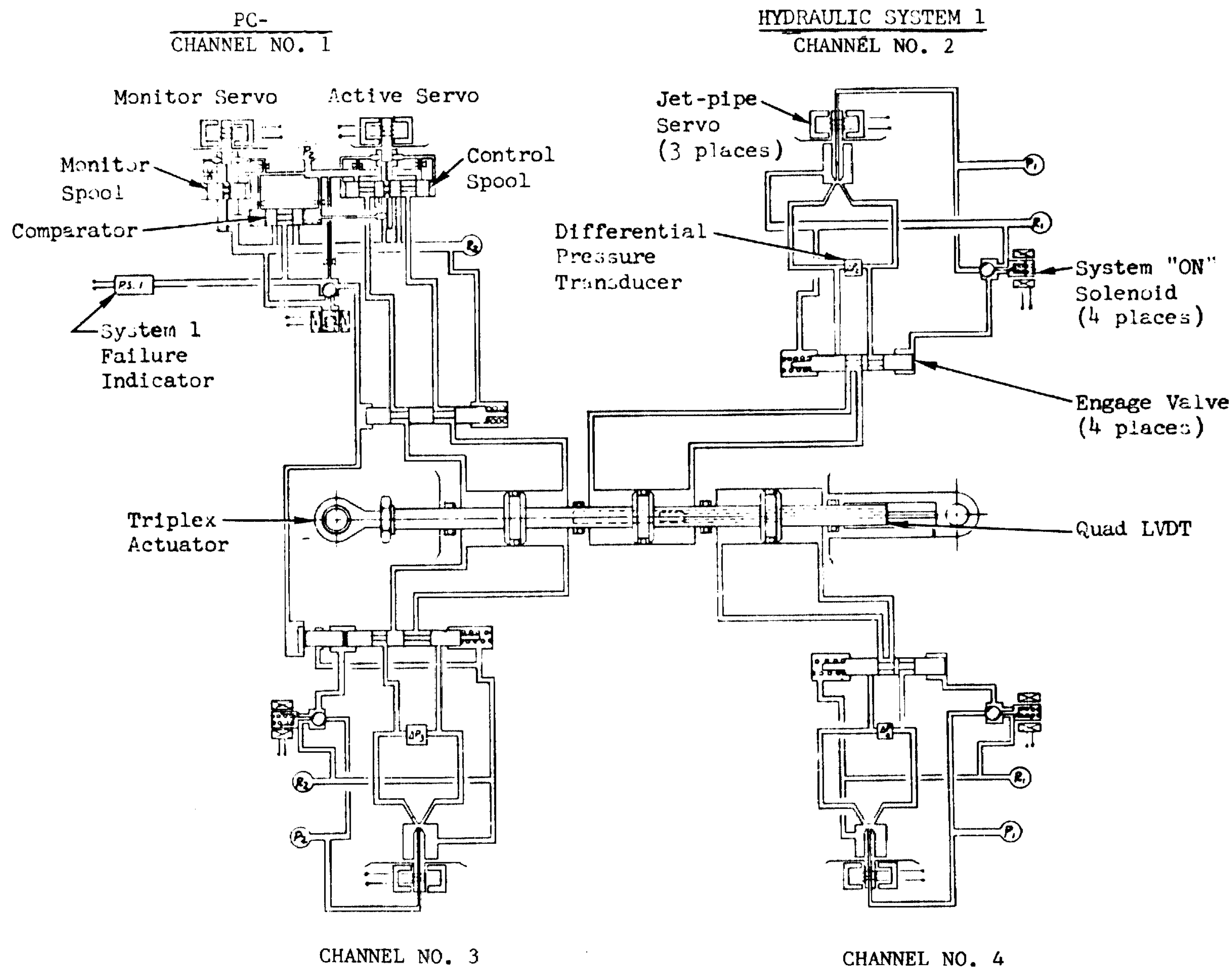
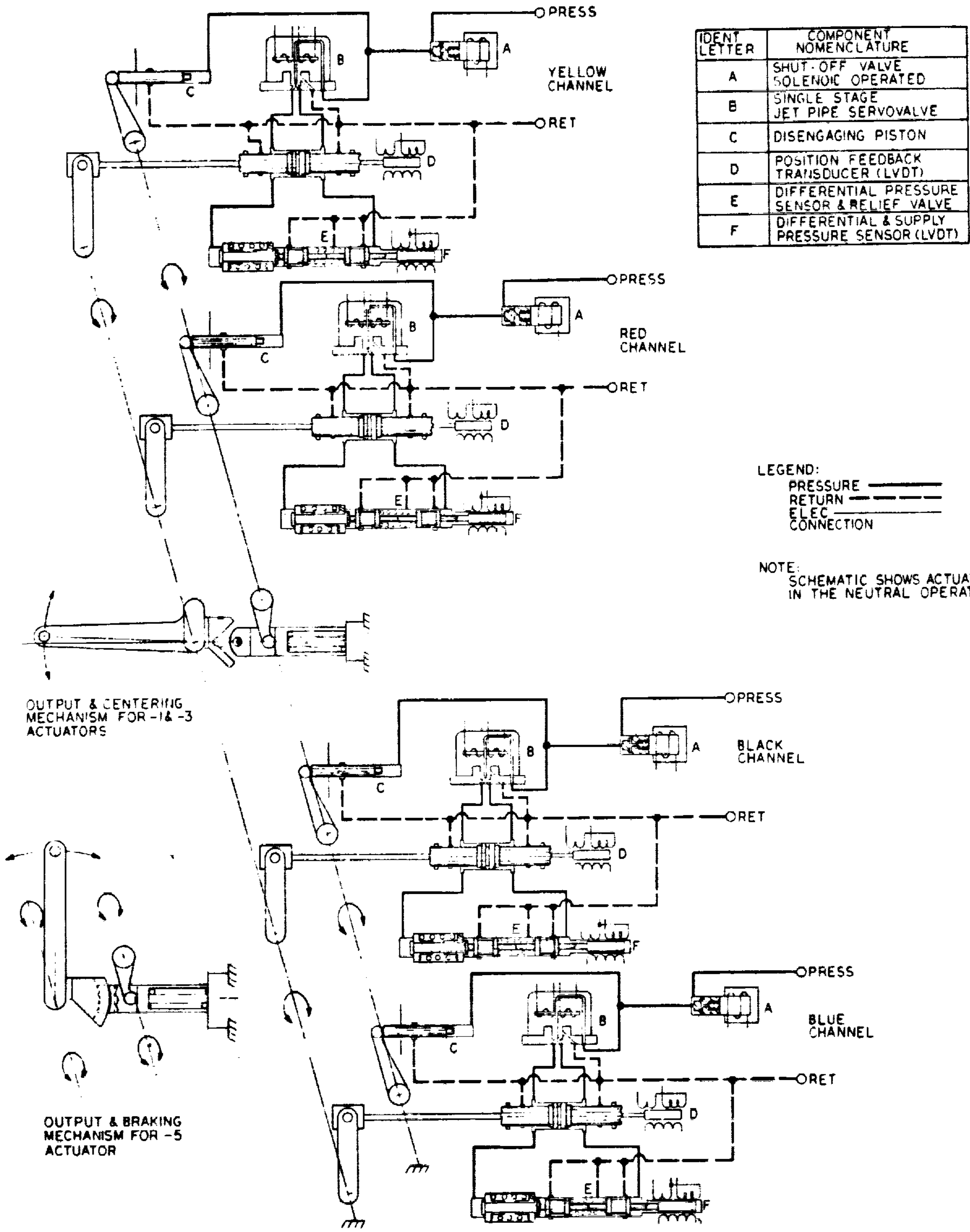


FIGURE 4.2-19 F-8 SECONDARY ACTUATOR SCHEMATIC



IDENT LETTER	COMPONENT NOMENCLATURE
A	SHUT-OFF VALVE SOLENOID OPERATED
B	SINGLE STAGE JET PIPE SERVOVALVE
C	DISENGAGING PISTON
D	POSITION FEEDBACK TRANSDUCER (LVDT)
E	DIFFERENTIAL PRESSURE SENSOR & RELIEF VALVE
F	DIFFERENTIAL & SUPPLY PRESSURE SENSOR (LVDT)

LEGEND:
PRESSURE
RETURN
ELEC CONNECTION

NOTE:
SCHEMATIC SHOWS ACTUATOR ENERGIZED (UNLOCKED) IN THE NEUTRAL OPERATING POSITION

FIGURE 4.2-20 680J SECONDARY ACTUATOR SCHEMATIC

TABLE 4.2-5 SECONDARY ACTUATOR DATA

ITEM	NASA F-8	680J
Piston Area	5.08 mm (0.2 in.)	7.5 mm (0.294 in.)
Stroke	50.8 mm (2.0 in.)	50.8 mm (2.0 in.)
Weight	.12 kN (26.25 lbs)	.17 kN (39.0 lbs)
Coulomb Friction		.07 kN (16 lbs)
Open Loop Gain		122
Maximum Piston ΔP		6.9 N/mm (1000 psi)
Servo Valve	2 Stage Flapper/ Noz	Single Stage Jet Pipe
Monitor	Hyd.	Electronic
Act. Centering Upon Total Failure	No	Yes

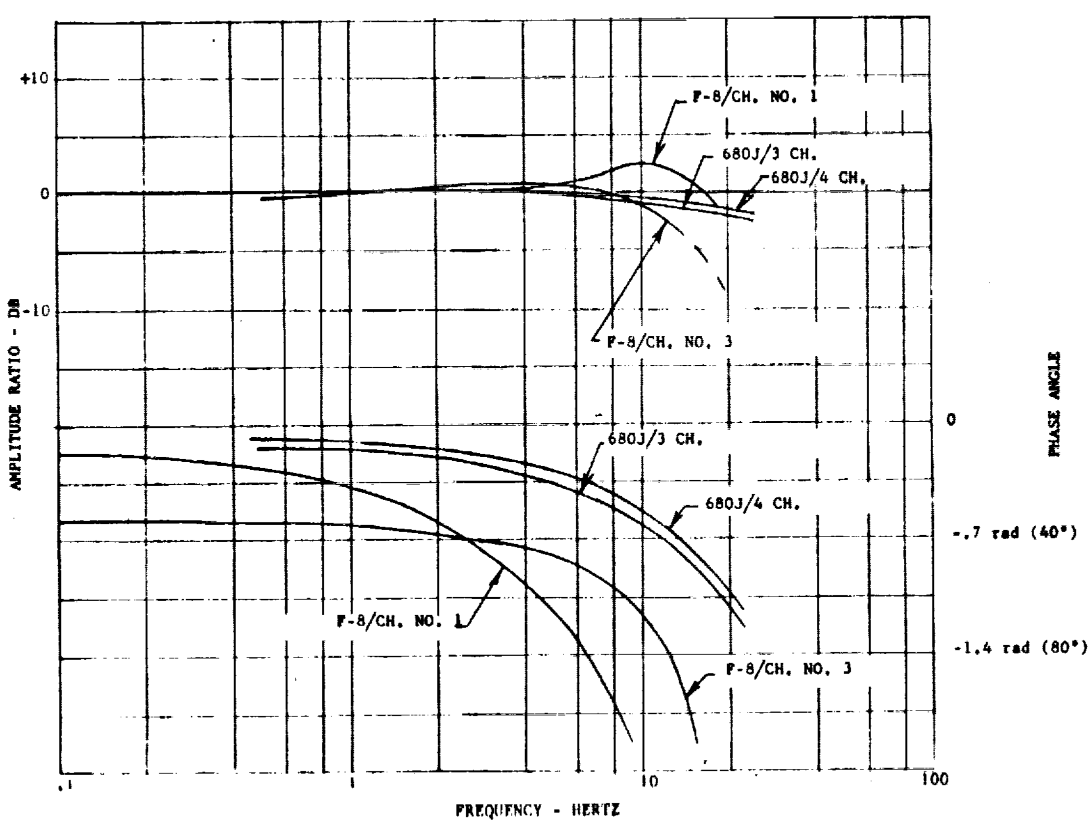


Figure 4.2-21 SECONDARY ACTUATOR RESPONSE

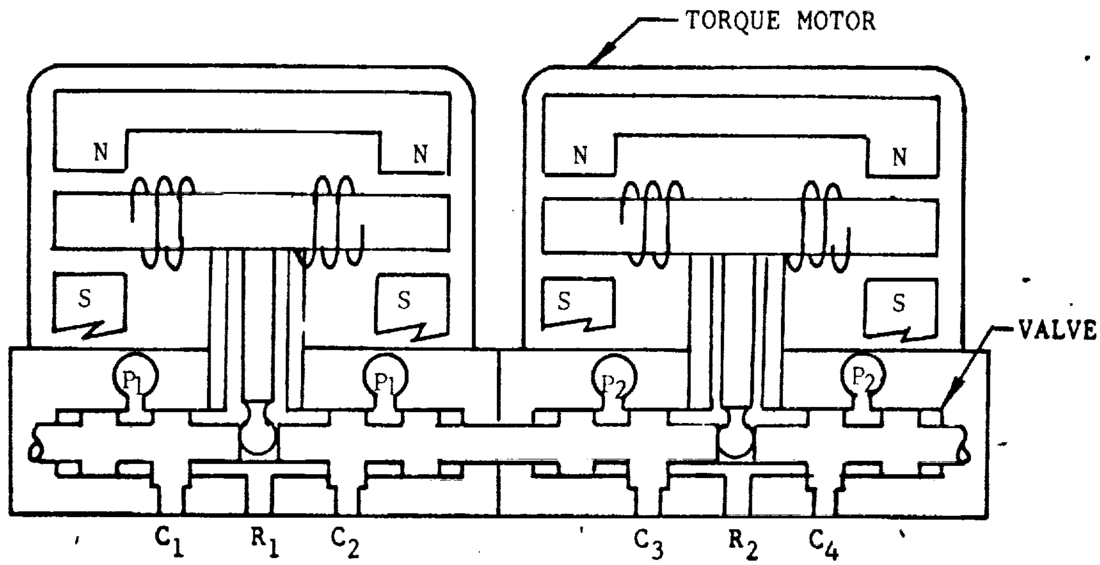


FIGURE 4.2-22 PM TORQUE MOTOR VALVE

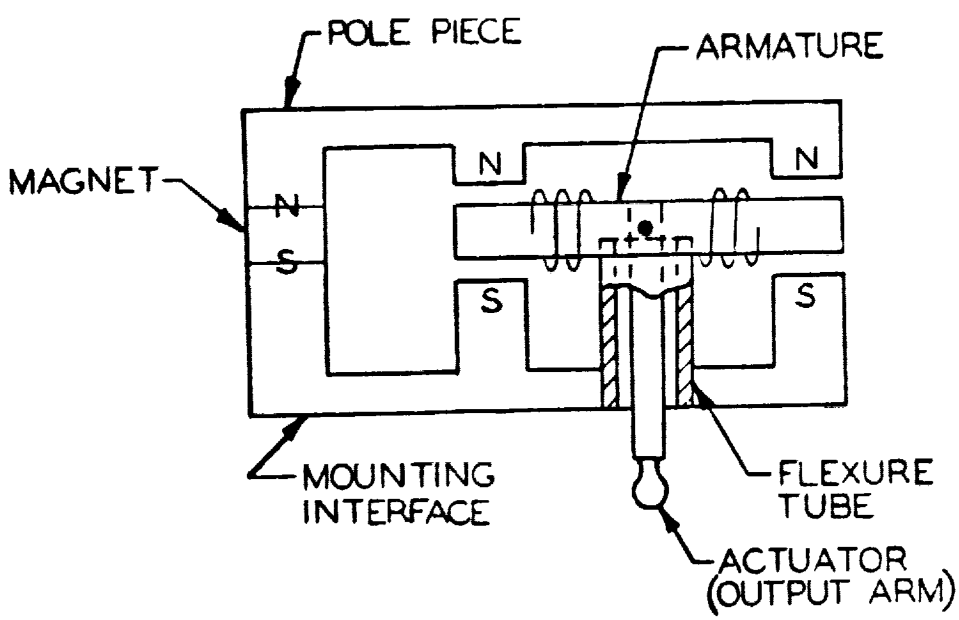


FIGURE 4.2-23 PERMANENT MAGNET TORQUE MOTOR

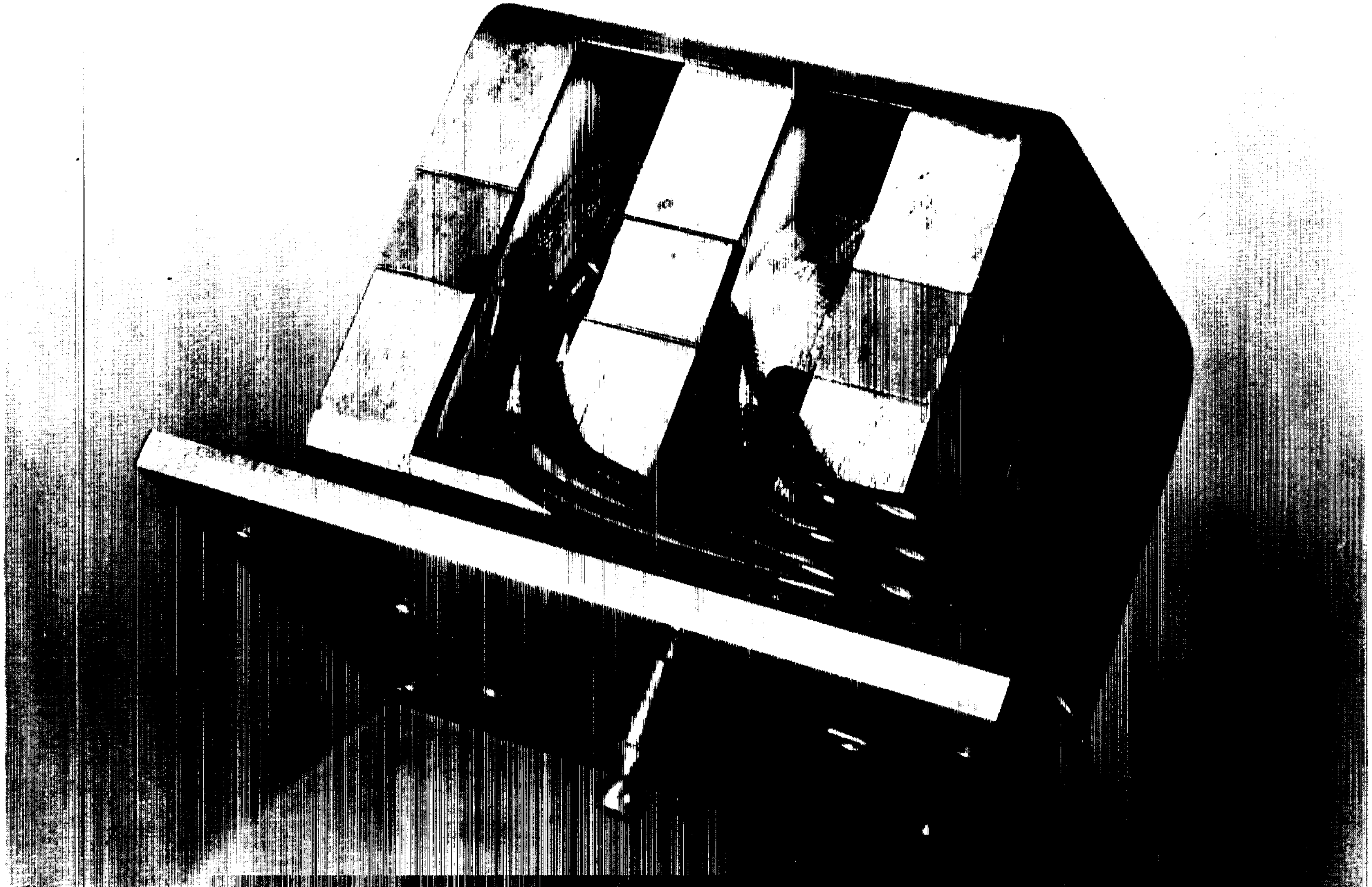
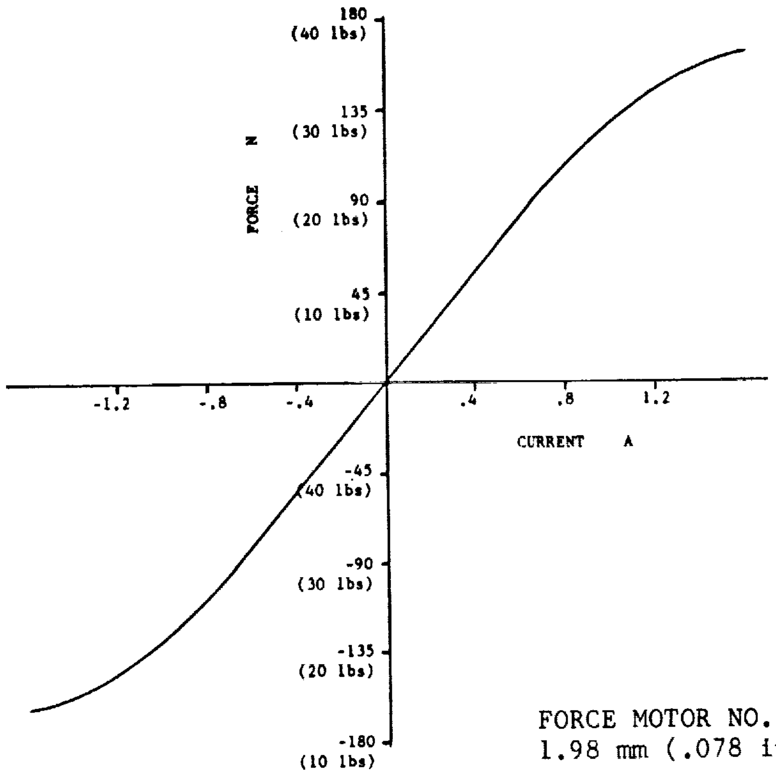


FIGURE 4.2-24 LABORATORY MODEL - PM TORQUE MOTOR



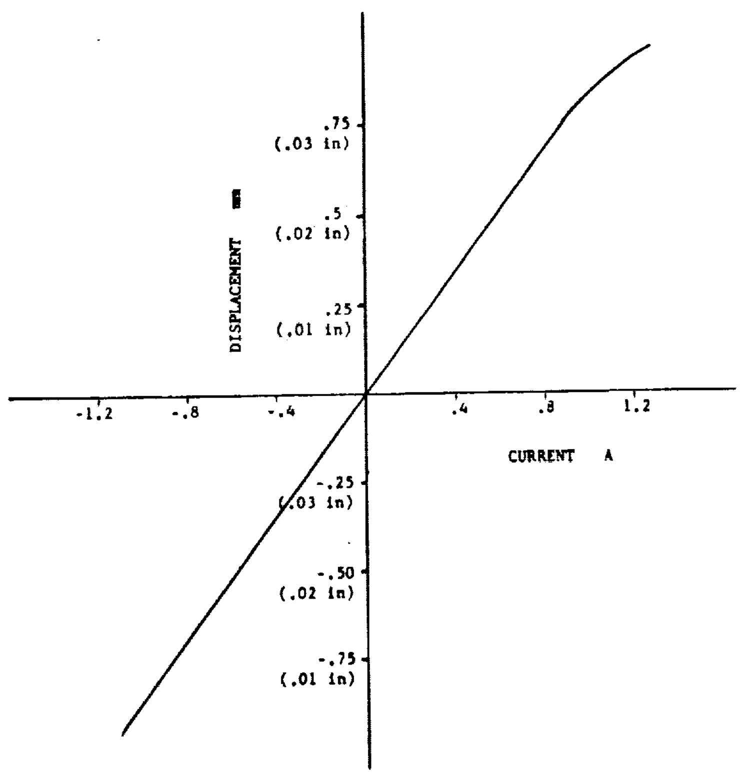


FIGURE 4.2-25 LABORATORY MODEL CHARACTERISTICS

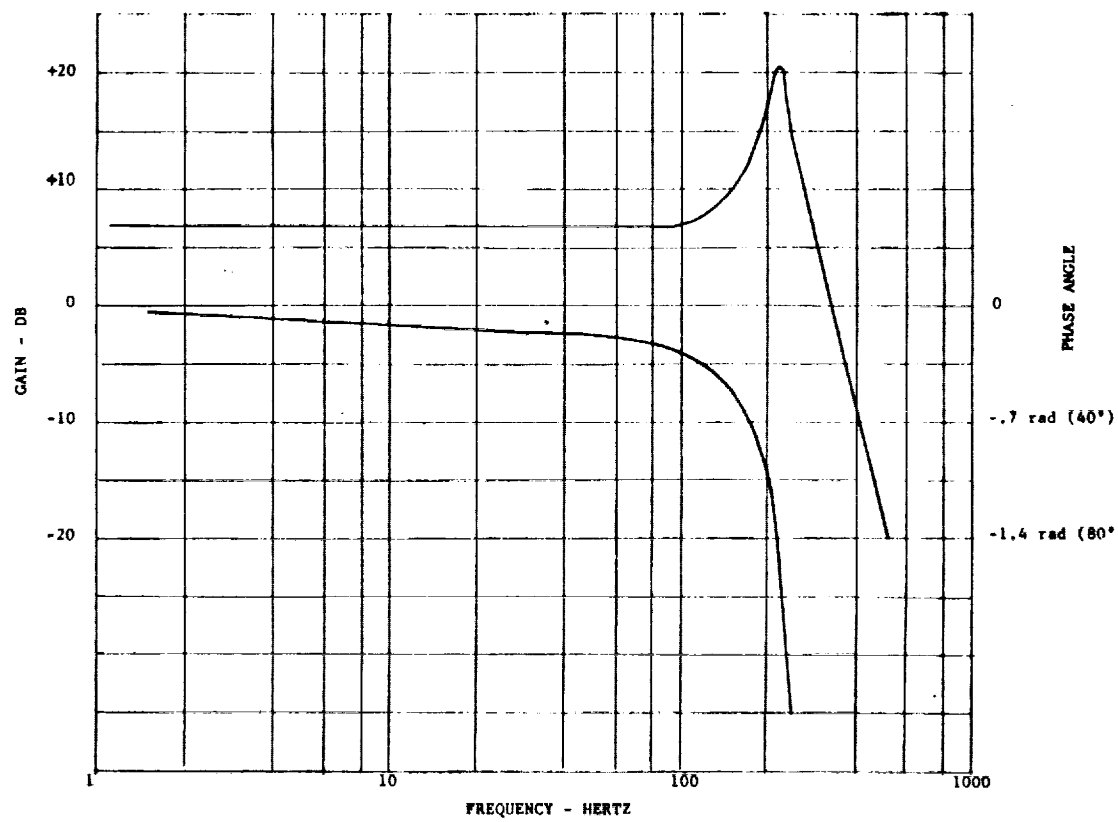


FIGURE 4.2-26 PM TORQUE MOTOR RESPONSE

8). The high energy product of colbalt samarium and its high coercive intensity permit compact, high-torque design. The material is very permanently magnetic, enabling operation with virtually no transient-induced demagnetization. Its high coercive strength permits large air gaps at high flux densities, a basic requirement of this force motor. A photograph of a laboratory model is shown on Figure 4.2-24. Salient features are high force output, long stroke, rugged, permanent magnet design, and redundant coils. Field flux is produced by six cobalt samarium permanent magnets. Armature windings are composed of four separate, independent coils made of AWG No. 25 copper wire.

Static and dynamic characteristics of this particular design are shown in Figures 4.2-25 and 4.2-26

In the stepper motor direct drive concept, dual stepper motors drive the power control valve through differentials as depicted in Figure 3.4-2.

The foremost characteristics required of a valve driver for use in a FBW system are reliability, shearout force capability and response. The stepper motor is a simple rugged reliable device and satisfies the first requirement. The stepper motor is capable, through suitable gearing, of producing adequate shear-out force. The stepper motor is limited in response due to its rate limit and acceleration time constant. The following calculations illustrates the latter two characteristics by applying a typical stepper as a valve driver for the stepper motor direct drive approach.

Experience indicates 0.71 kN (160 lbs) shear-out force capability as a reasonable compromise for FBW applications. Each motor must produce half or 0.36 kN (80 lbs) at the valve spool. This requirement establishes the gearing (K_g) between the stepper motor and the valve spool. Assuming 50% torque efficiency in the gearing then:

$$50\% \times \Upsilon \times K_g = .036 \text{ kN (80 lbs)}$$

$$\text{or} \qquad K_g = \frac{80}{.5 \times 2.6} = 2.43 \text{ mm}^{-1} \text{ (61.6 in}^{-1}\text{)}$$

The maximum actuator rate produced by this gearing for the selected stepper motor [314 rad/s (3000 rpm)] is:

$$\dot{X}_{max} = \frac{2 \times 3000}{61.6 \times 60} = 129.5 \text{ mm/s (5.1 in/sec)}$$

Considering a sinusoidal input the maximum rate can be shown to equal:

$$\dot{X}_{max} = A \times (\omega) \qquad \text{where} \quad A = \text{amplitude}, \quad \omega = \text{frequency}$$

Assuming $\pm$101.6 mm ($\pm$4 in) maximum actuator stroke as a design point then:

$$\omega = \frac{\dot{X}_{max}}{A} = \frac{5.1}{20\% \times 4.0} = 6.37 \text{ rad/s}$$

The actuator would rate limit at this frequency, obviously far below the required 13.5 rad/s specified for high performance aircraft.

One method of increasing the maximum rate is to utilize a two stage stepper motor driven valve design (ref. 166). The two stage design utilizes hydraulic pressure to develop the shear out force. It is capable of providing very high shear-out force. Shear-out force is determined by the sizing of piston area within the valve and not upon the stepper motor output torque. In this design the stepper motor must only overcome frictional loads in positioning the pilot spool. For purpose of discussion assume the gearing is changed to 1.2 mm^{-1} (30 in^{-1}) then the force available at the pilot spool is:

$$F = .5\ (2.6)\ (30) = 0.17 \text{ kN } (39 \text{ lbs})$$

$$\dot{X}_{max} = \frac{2\ (3000)}{30\ (60)} = 264 \text{ mm/s } (10.4 \text{ in/sec})$$

which is fast enough for the F-8 ailerons. The frequency at which rate limiting would occur is:

$$\omega = \frac{10.4}{.2(4)} = 13. \text{ rad/s}$$

which is above the F-8 passband and nearly equals the specified high performance requirements. The quantization step size is:

$$q = \frac{2.25}{57.3 \times 30} = .033 \text{ mm } (.0013 \text{ in})$$

4.2.3.3 Interface Electronics. The interface electronics to couple the PM torque motor actuation system to the three and four channel digital FBW systems are depicted in Figures 4.2-27a and 4.2-27b. Salient features of the design are the dual-quad or dual-triple adaptive gain valve driver amplifier, redundant D/A conversion and feedback demodulator circuits, and the simple Backup Control System interface.

This interface accepts computer processed signals and uses standardized modules which can be assembled using the building block approach to provide the desired configuration. Two torque motors directly drive the dual tandem valve/actuator used in system. Each motor has redundant windings, with each winding requiring a maximum drive power of 4 watts at maximum input command.

High reliability is achieved in that multiple sub-unit failures can be accommodated before the gain ratio, motor current per unit of input control signal voltage, is appreciably affected. This is accomplished by feeding back a common point voltage proportional to the total motor current to the input of each sub-unit.

High loop gains needed in an actual mechanization require tight control over the dc offset differentials. This does not pose a serious problem but must be accounted for in the design. Of greater significance is the effect of the relatively high gains of the amplifiers on the static motor currents due to differential offset voltages between feedback signals and between input signals. This can be seen by calculating the error signal needed at the valve driver input to produce full rated motor current. In a system where ±12 vdc represents full scale command signal input and actuator feedback voltage levels, high gains (60 amp/volt or more) would mean that total static errors equal to or greater than about 0.25% to 0.5% of full scale would result in excessive static motor currents. Some potential sources of offset differentials are static scale factor and tracking differences in the feedback and input signals; null differentials and/or dc offsets in the amplifiers, feedback and input signals; and inaccuracies in the summing circuits. The major potential error source is the difference in regulation of the ac reference voltages used by the various feedback and stick LVDT's.

At least two approaches are possible to control offset errors. In one approach, "cross-feeding" the ac references between channels could minimize the ac regulation error source; inductive summing of stick and feedback signals prior to demodulation would further reduce tracking errors as well as input summing errors; and precision scaling circuit sources. (The computer-derived surface commands could be forced to match precisely, so that input signal errors would not exist in the digital flight control modes). Further study would be required to determine (a) the cross-fed connections and inductive summing arrangements needed to sum the several stick and feedback signals available at each surface from redundant paths, and (b) the tolerance of this arrangement to single failures that might otherwise upset the dual balance.

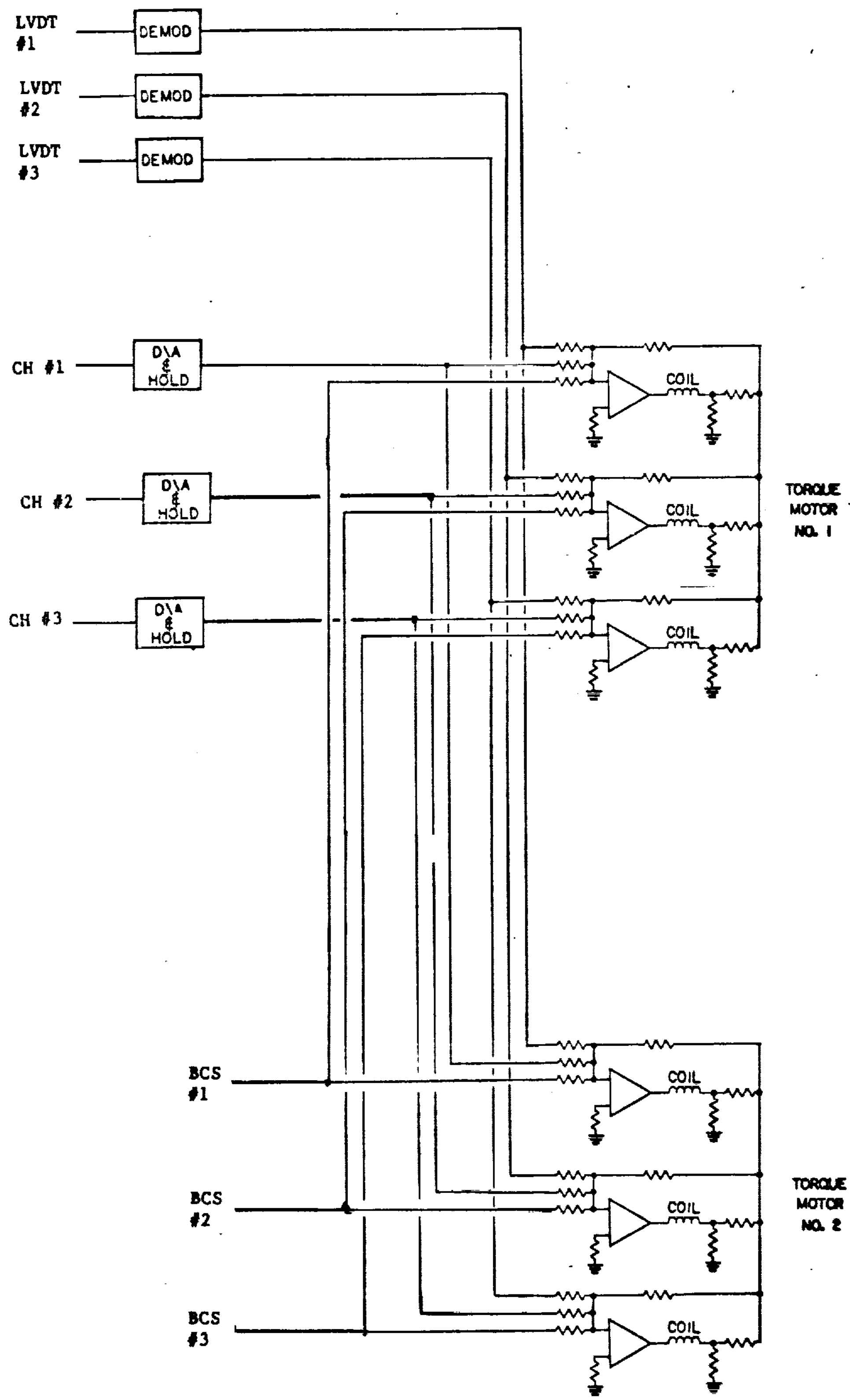


FIGURE 4.2-27a PM TORQUE MOTOR - 3 CHANNEL INTERFACE

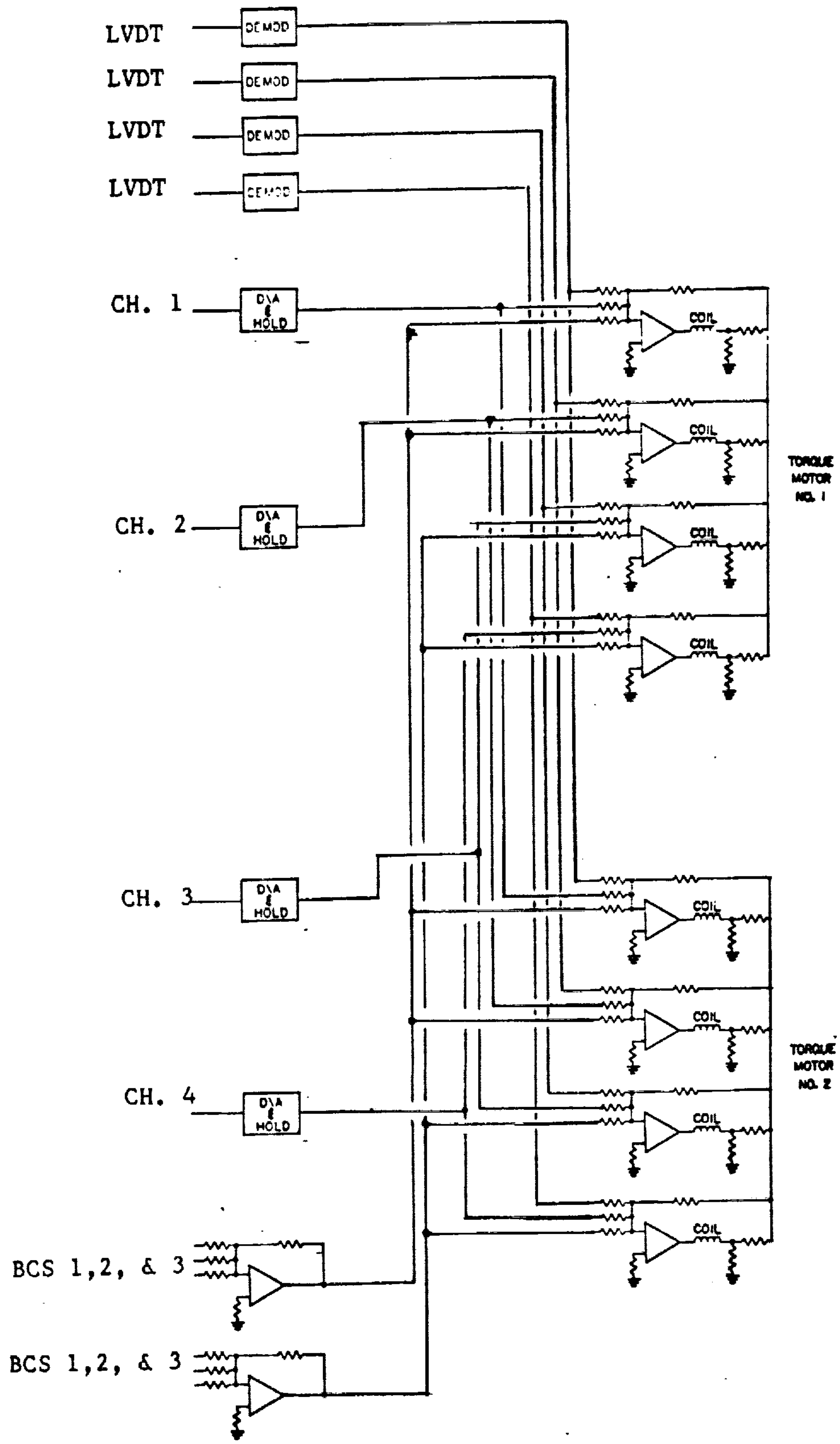


FIGURE 4.2-27b PM TORQUE MOTOR - 4 CHANNEL INTERFACE

A second approach available that makes the dual drive amplifiers relatively insensitive to static errors is to feed each computer output and each feedback signal into both of the redundant valve drive amplifiers. Input offsets are thereby balanced. It is still necessary to control the DC offsets within each valve driver subunit. Electrical redundancy is maintained completely throughout the design. This latter approach is depicted in the selected configuration.

The "D/A & Hold" data converter circuit consists of a series to parallel signal converter, a buffer register and a D/A converter. The actuator command, in serial digital word format, is sent to the data converter circuit along with a clock pulse. The serial format is converted to parallel format and held in a buffer register until the succeeding sample period at which time the register is updated. Output of the register feeds a parallel D/A converter circuit producing an analog signal command.

The baseline mechanization established offers inherent compatibility with a conceptual computer mechanization for three and four channels. In essence two parallel highly reliable, servo amplifiers are used. Each of the dual amplifiers is quad redundant. It would be necessary for at least five amplifier subcircuits to fail hardover in the same direction before a system hardover will occur. The probability of such a concentration of failures is highly remote.

In summary the PM torque motor direct drive actuation system concept is directly compatible with the projected digital flight control computer system, regardless of the ultimate degree of redundancy. The design details will offer several tradeoff areas, but these must await a firming up of the digital computer hardware and detailed interface circuit design.

The three and four channel interface electronics for the secondary actuator approach is depicted in Figure 4.2-28a and 4.2-28b. The amplifier design concept parallels the PM torque motor amplifier design. Dual coils of the jet pipe servo valve are driven by current amplifiers with common point feedback. The two halves of the valve driver compliment each other in a manner such that if the output of one drops off the other increases to compensate; should one fail hardover the other is driven hardover in the opposite direction thereby cancelling its effect. The amplifier design essentially fails passive regardless of amplifier component failures. In the three channel system a backup control system input is fed directly to each channel. In the 4 channel system a backup control system input is fed separately to three channels and the sum of the three backup control system inputs is fed to the fourth channel. The "D/A & Hold" data conversion circuit is similar to that used in the PM torque motor interface circuit.

The basic stepper motor drive electronics is depicted in block diagram form in Figure 4.2-29. The system consists of an input register, a comparator, a pulse generator, an up-down counter, and a power switch unit.

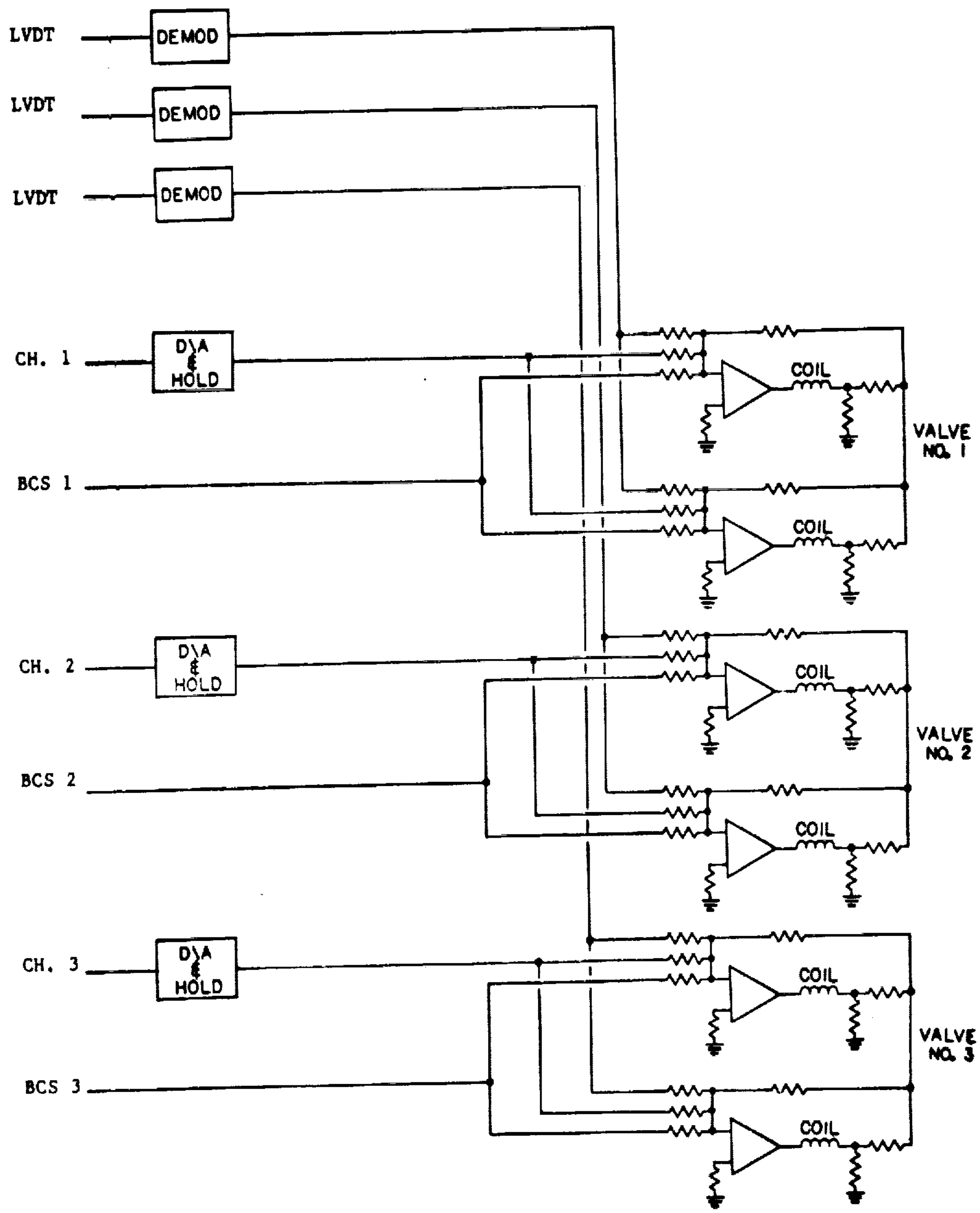


FIGURE 4.2-28a SECONDARY ACTUATOR-3 CHANNEL INTERFACE

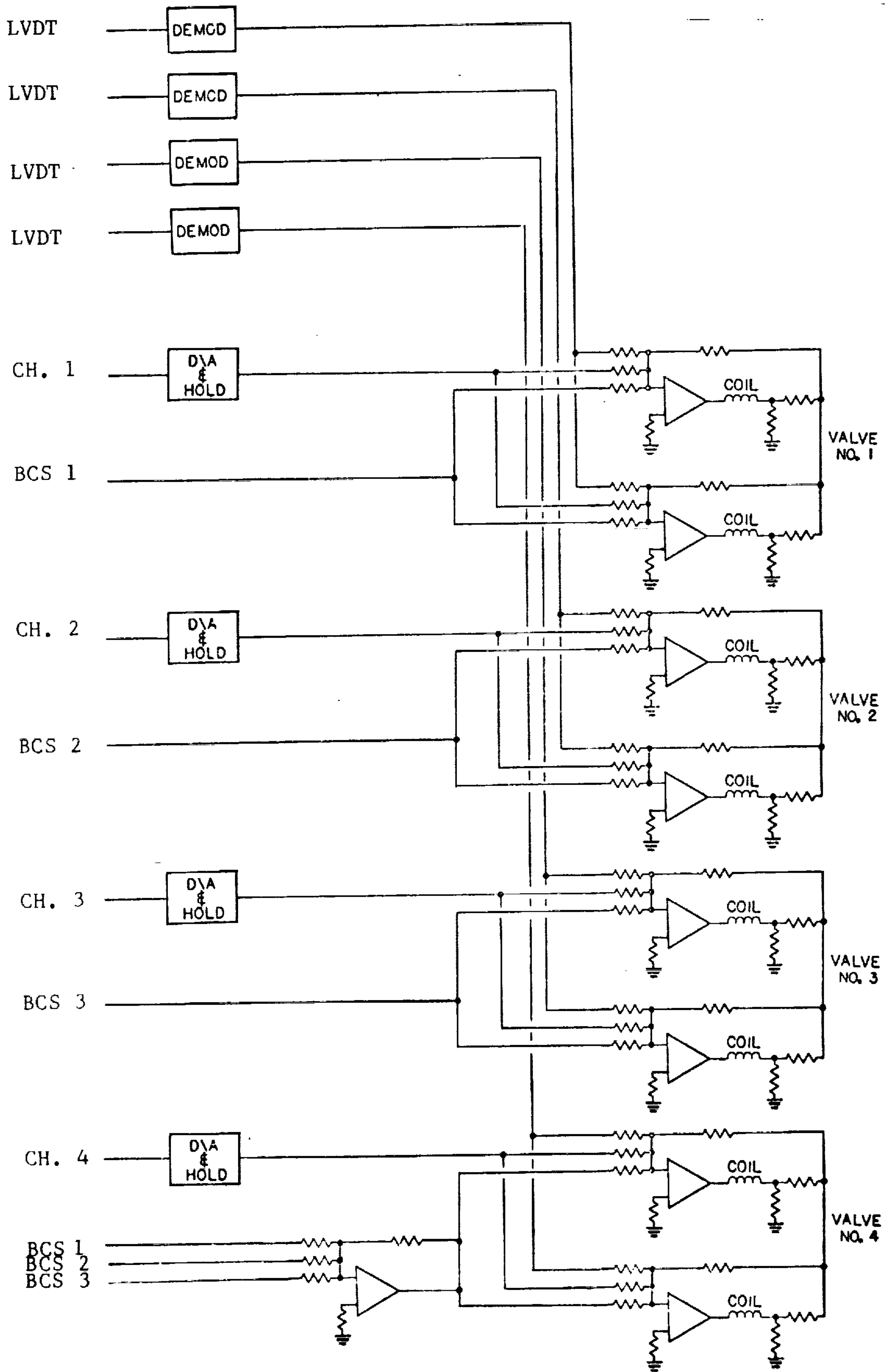


FIGURE 4.2-28b SECONDARY ACTUATOR-4 CHANNEL INTERFACE

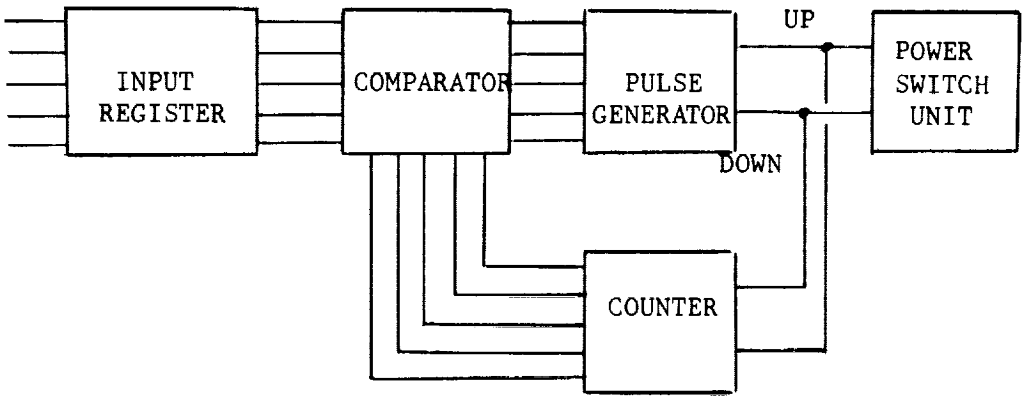


FIGURE 4.2-29 STEPPER MOTOR DRIVE

The parallel word input command is transmitted to and stored in the input register at each sample period. The input register holds the commanded position between samples. The commanded position and the actual position are compared in the comparator, the difference information of which is transmitted to the pulse generator. The pulse generator produces the pulse train required to move the stepper motor to the commanded position in minimum time. If the required steps are less than the start-stop capability, a pulse train of the correct number of pulses at the start-stop rate is sent to the power switch unit. The up-down counter counts the pulses transmitted to the switching unit thereby keeping track of the stepper motor position. The power switching unit energizes the proper phase coil in sequence to execute the specified steps. At the next sample period a new comparison is made and the sequence repeats. Should the stepper be unable to reach the commanded position within the sample period, the error signal will increase. The pulse generator generates a time optimal pulse train variable in frequency and number to minimize position error. Pulse rates are never allowed to exceed the synchronous capability of the stepper motor. The phase switch unit excites the proper phase coil sequence to drive the proper motor.

Each successive input pulse alternatively turns on or turns off the current to a motor coil with either two or three coils energized at all times. The proper switching sequence for the motors is shown in Figure 4.2-30, indicates the coil is energized.

	STATE									
	1	2	3	4	5	6	7	8	9	10
Phase A	1	0	0	0	0	0	1	1	1	1
Phase B	1	1	1	0	0	0	0	0	1	1
Phase C	1	1	1	1	1	0	0	0	0	0
Phase D	0	0	1	1	1	1	1	0	0	0
Phase E	0	0	0	0	1	1	1	1	1	0

← Reverse Forward →

FIGURE 4.2-30 Switching Sequence

Ten coil excitation states exists. After ten successive steps in the same direction the sequence repeats. The power switch unit drives each coil via a solid state power switch utilizing two supply voltages to excite the phase windings, a high voltage to overcome winding inductance effects in delaying current build-up and a low voltage to maintain the required coil current once it has reached the specified level. Figure 4.2-31 depicts the principal of this approach applied in a redundant circuit.

The complete dual redundant stepper motor drive circuit is depicted in Figure 4.2-32. It is completely redundant except for the phase coils.

Backup control system commands (analog signals) must be converted to digital commands to be accepted by the stepper motor drive circuitry. The simplest method of interfacing with the backup system is to convert each of the three signals to the proper digital format and feed them into the signal selectors as depicted in Figure 4.2-31. The BCS Control Signal informs the signal selector circuit when to accept BCS commands in lieu of digital flight control computer signals.

The entire drive circuitry including the BCS section can be LSI CMOS circuitry except for the power switching transistors.

4.2.3.4 <u>Loop Closure</u>. The advantages of closing the servo loops about the digital computer are; flexibility, versatility, noise insensitivity, compensation by non-linear programming and adaptive or self-optimizing control, time sharing, and self reorganizing to name a few. Digital or sampled data control loops have been extensively investigated and utilized during the past two decades. The disadvantages are the increase in computer size and the added complexity in the backup control system interface. Actuation servo

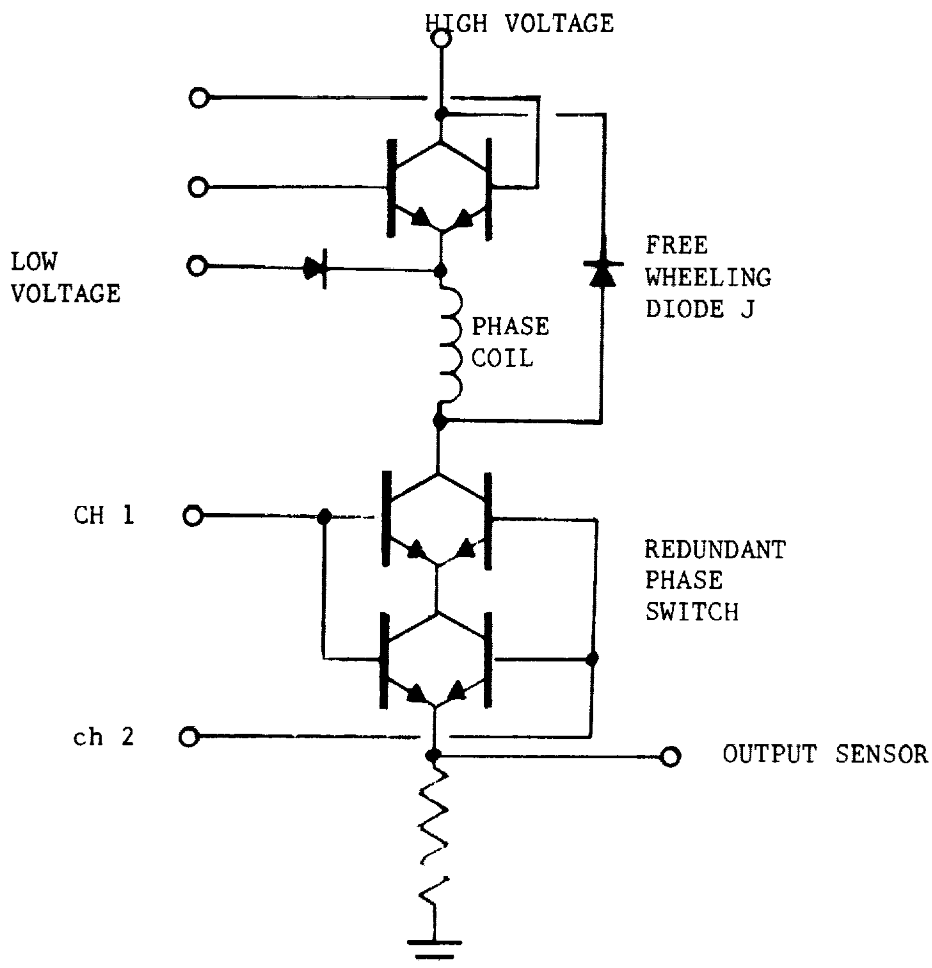


FIGURE 4.2-31 PHASE COIL SWITCH

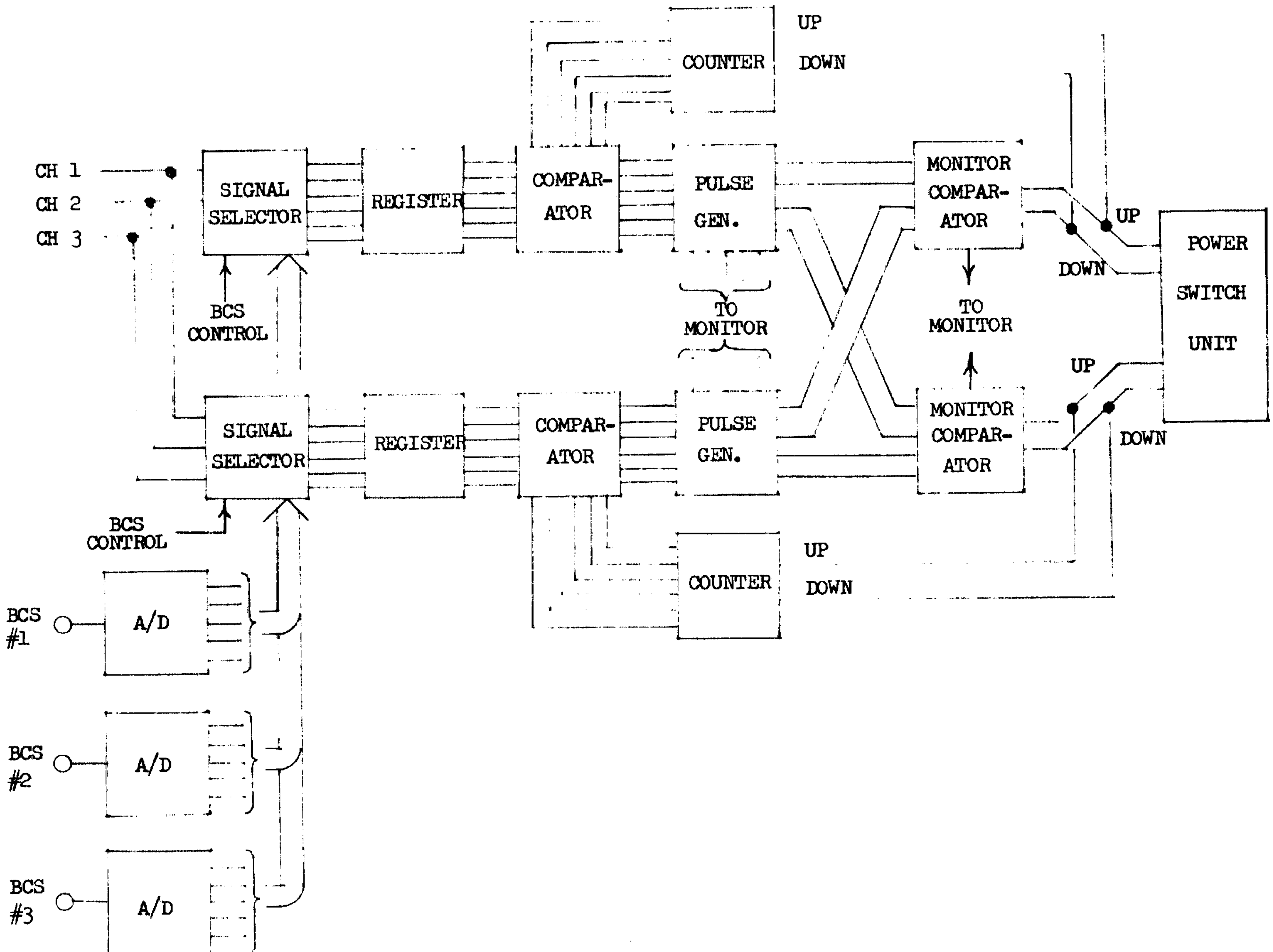


FIGURE 4.2-32 REDUNDANT STEPPER MOTOR DRIVE

loops operate at relatively high frequencies (i.e., 20 hertz), consequently high sampling rates are required. The three axis FBW system being considered utilizes five power actuators which may, due to redundancy, require as many as 20 servo loops. This computational load increases the required size of the computer. Most flight control computer designers agree that with the present state of technology (Honeywell for DIGIFLIC, Sperry of AFTI, etc.) closure of the servo loops outside the digital computer permits a more optimum design from size, weight, and power considerations. In order to be competitive with an analog FBW mechanization, a two to one improvement in the physical characteristics and power requirements of digital computers is required. CMOS technology holds promise, of such an improvement in the future, but is not available at this time.

The three concepts studied have, for the above reasons, closed the loops outside the digital computer, two electrically and one mechanically. However recognizing that there may be some other reason for closing the loop about the computer a cursory look at the three concepts was made to see if any one approach had significant advantage over the other. No one approach appeared to have a significant advantage over the others.

4.2.3.5 <u>In-Flight Monitor</u>. Design of an in-flight monitor depends entirely upon the total system design. To illustrate, the reduction in the catastrophic failure probability of the four channel PM torque motor approach realizable by cross channel in-flight monitor is 0.01%. On the other hand, in a flight control system where the only monitoring accomplished is channel comparison in the actuation system the improvement realizable by the monitor (assuming total channel failure probability $Q_A = 1000 \times 10^{-6}$) for a four channel system is 40%. The effect of an in-flight monitor in the secondary actuator approach is relatively the same as for the PM torque motor approach. The stepper motor approach is inherently dependent upon the monitor regardless of the total flight control system design approach.

Figure 4.2-33 presents a block diagram of the secondary actuator in-flight monitor circuit utilized in the 680J design (142). The diagram shows that the signals from the differential pressure (ΔP) LVDT are demodulated and compared in six cross channel comparators whose outputs are connected to logic circuits for failure detection and shutdown of a failed element. Each channel is equipped with a pressure transducer (LVDT) connected across each secondary actuator piston. The transducer senses the absence or low system pressure, and excessively high differential piston pressures. A carrier sensor circuit is incorporated which will detect an LVDT open, short or the power failure. The monitor time delay for the comparators is 0.5 second.

A hardover type failure is easily isolated by the circuit. A hardover failure in the blue element, for example, produces three signals to the AND gate in the blue element logic circuit approximately 0.5 seconds later, and results in a shutoff signal to the blue element shutoff valve. A single passive failure, on the other hand, will not be detected under normal operating conditions because the failed element does not generate enough differential pressure to

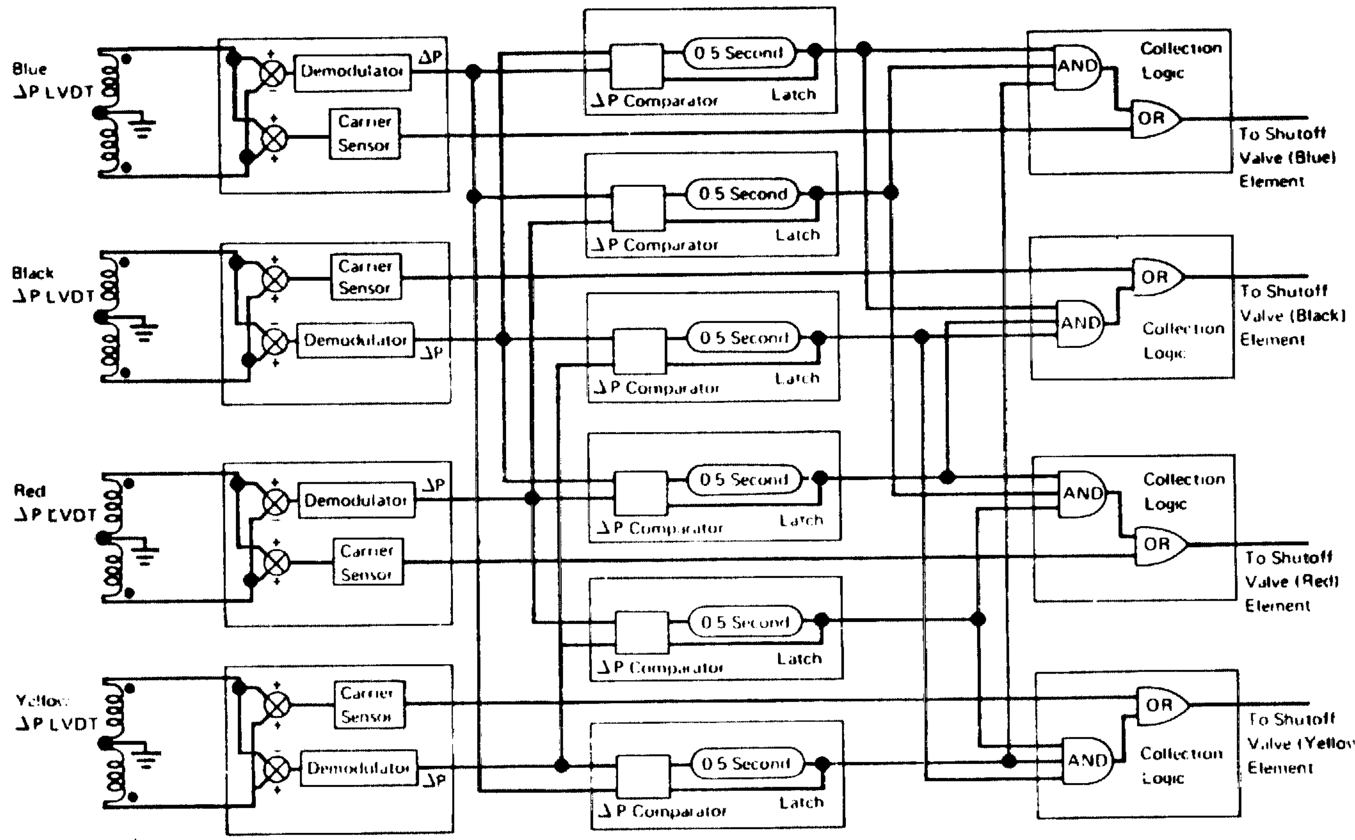


FIGURE 4.2-33 SECONDARY ACTUATOR IN-FLIGHT MONITOR

produce failure indication. Detection and fault isolation will occur under stall conditions, however, since all three operating elements will generate failure signals. When two passive failures have occurred, switchout of either passively failed element can not be accomplished even under stall conditions because it is no longer possible to produce three signals to an AND gate.

The operation of the in-flight monitor circuitry is considered to be satisfactory since the presence of one or two passive failures does not degrade the secondary actuator performance appreciably. In addition, the exercise of BIT before each flight will detect the existence of passive failures at that time.

A block diagram of a monitor for the PM torque motor approach is depicted in Figure 4.2-34. The basic parameter monitored is torque motor coil current. Current is sensed by the small series resistor used in the feedback design of the current amplifier. Each coil current is differentially summed with the average motor currents. The difference is fed to a threshold detection and holding circuit. This circuit produces a logical 1 indication when the differences exceed a specific threshold for a specific period of time. Once in the logical 1 state the circuit latches. The output of the threshold circuit is fed to an AND gate along with the error signal generated by the complimentary coil current error detection circuit. For example, the computer channel no. 1 signal drives coils no. 1 and no. 5, thus the error indications from no. 1 and no. 5 threshold circuits are fed to an AND gate. The AND gates produce an indication of a computer channel failure or an input failure to the actuation system.

The monitor provides indication of both torque motor amplifier failures and system channel failures. Hardover type failures are readily detected by this circuit. Detection of open type failures depend upon the delay time and threshold level utilized in the threshold circuit. An error indication is generated only when a servo error above the threshold level exists for a period greater than the delay time (when actuator rate is commanded). The design must compromise between low thresholds, short delays, and nuisance indications. Opens produce the least degradation in actuation system performance of any failure. Internally the actuation system can operate with at least four open failures without loss of performance. Externally the digital computer monitoring would be expected to sense input opens. The monitor is very simple requiring only a few CMOS integrated circuits, it does not require any additional sensors such as LVDT's, pressure transducers, etc.

The stepper motor monitor is an integral feature of the drive circuitry, and is depicted in Figure 4.2-32 and 4.2-35. The monitor compares the operation at the pulse generator output by comparing each pulse being sent to the power switch unit. Should disagreement exist the monitor halts normal operation and the state of each drive path is interrogated by comparing the number of pulses remaining in each pulse generator to be transmitted to the power switching unit. The monitor resolves which path is in error by comparing the commanded state at the pulse generator output with the computer inputs. The faulty unit is disabled and the functional unit is permitted to continue. Cross monitoring between the stepper motor channels is also performed to provide a further check upon the operation and to enable isolation of faulty units upon additional failures.

Another approach to monitoring of redundant actuation system channels is in-line monitoring. In this approach the health of each channel is determined entirely upon information within the channel, without reference to any other channel. This technique is very effective in purely electronic systems and somewhat difficult to apply in systems with electromechanical and hydro-mechanical components.

The monitoring to be incorporated into the actuation system is dictated by the total flight control system design concept. All approaches lend themselves to fairly simple cross channel comparison monitor schemes. Cross channel comparison monitor techniques are of questionable value for three channel systems; it does not improve the probability of having control but it does improve the probability of not having a hardover. The better approach to monitoring is in-line monitoring. In-line monitoring is more difficult to apply in systems with mechanical components. The PM torque motor approach has less electromechanical and hydromechanical components and therefore lends itself more readily to in-line monitoring techniques.

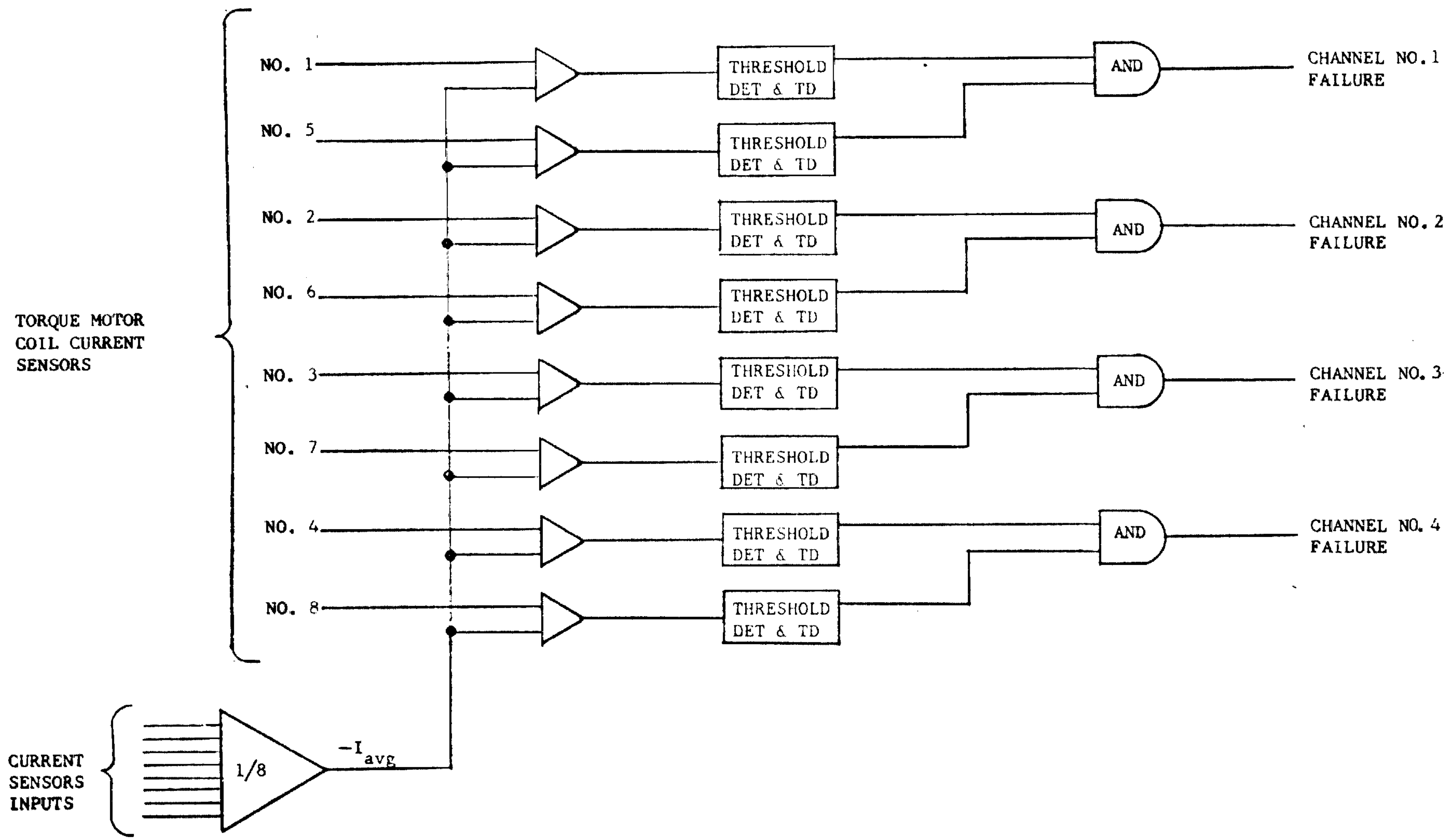


FIGURE 4.2-34 PM TORQUE MOTOR APPROACH MONITOR

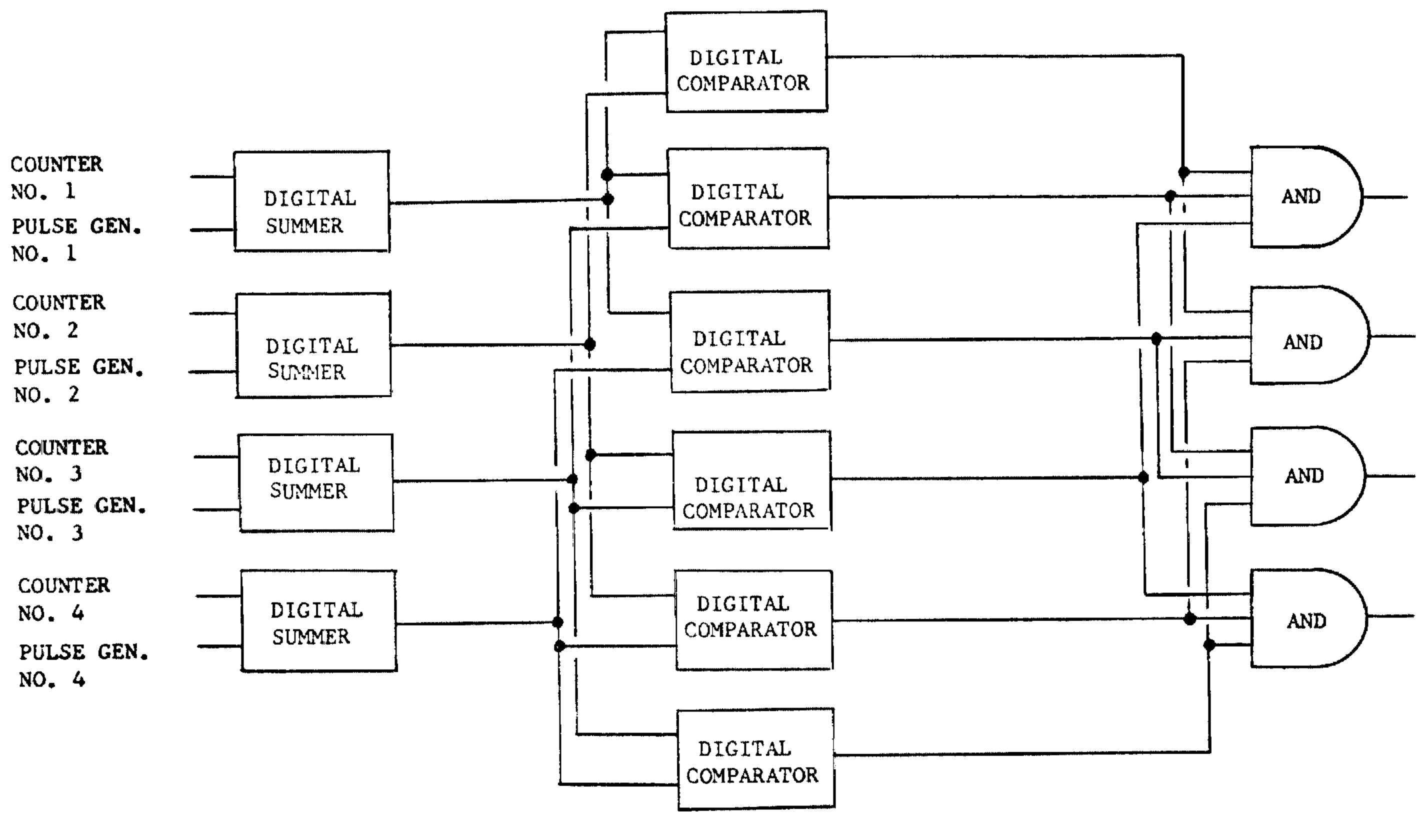


FIGURE 4.2-35 STEPPER MOTOR MONITOR

4.2.4 POWER SYSTEMS IMPACT

The power consumed by each approach can be compared on a quiescent basis or on the basis of a specific input. Since the power actuator consumes a large amount of power for reasonable inputs and since the time average surface activity is generally quite small, the quiescent power is the best basis for comparison. Estimated quiescent power consumption of the three approaches are presented in Table 4.2-6.

TABLE 4.2-6 QUIESCENT POWER CONSUMPTION

	Electrical/Act.		Hydraulic/Act.		Total/Act.		Total/Aircraft	
	3 Ch.	4 Ch.	3 Ch.	4 Ch.	3 Ch.	4 Ch.	3 Ch.	4 Ch.
Secondary Act. Approach	64w	85w	36 cm^3/s (.58 gpm)	48 cm^3/s (.76 gpm)	882w	1075w	4120w	4500w
PM Torque Motor Approach	1.8w	2.4w	2.5 cm^3/s	2.5 cm^3/s	54w	55w	270w	275w
Stepper Motor Approach	9.8w	9.8w	2.5 cm^3/s (.04 gpm)	2.5 cm^3/s (.04 gpm)	62w	62w	310w	310w

The table contains both electrical and hydraulic power consumed in addition to the sum in watts. The secondary actuator, due primarily to solenoid shutoff valves consumes the most electrical power. The stepper motor approach is next due to the quiescent field coil current required. The PM torque motor consumes the least electrical power. The secondary actuator approach wastes a considerable amount of hydraulic power due to the quiescent flow required by the servo valves. The PM torque motor and stepper motor approaches suffer only from the main power spool valve leakage. The relative significance of the electrical and hydraulic power can be seen in the total power column. The hydraulic power losses completely over shadow the electrical losses. The PM torque motor and the stepper motor approaches offer a considerable power savings over the secondary actuator approach. The savings in total power. on an aircraft basis (5 actuators), is 5.1 kW for a 4 channel system of which 3.7 kW (5 hp) is hydraulic power.

4.2.5 Ecomonic Factors

Three economic factors were considered in comparing the three approaches: engineering costs, program costs, and cost minimization. The PM torque motor approach is the most economical to apply to the Phase II aircraft to provide the high performance desired for advanced and active control technology. The use of the existing secondary actuators is of course the cheapest since they are already designed, developed, and installed. This approach hinges upon whether the existing secondary actuators provide adequate performance for the intended use. The performance is certainly inadequate for the high performance requirements specified in Section 3. They could possibly be usable, depending upon the total flight control system design and the control laws which are to be investigated in the future test program. It is assumed for the purpose of this comparison that they would be replaced by higher performance units.

a. Engineering Cost

The PM torque motor design is the most economical approach in obtaining truly high response necessary for advance and active control configured vehicles. It is inherently capable of higher performance than either the secondary actuator or stepper motor approaches, therefore, constitutes a simpler easier design task. It is simpler, incorporating fewer parts, therefore, requiring less design effort. It is independent of monitors, therefore, eliminating a considerable design and development task. For low response applications, the secondary actuator approach is the most economical approach for the Phase II aircraft. It requires the least modification to the aircarft of the three approaches. It enjoys a high state of development and verification. The two direct drive approaches both require some development, primarily in the electronic drive design. A PM torque motor direct drive valve itself has been designed, fabricated, and is currently undergoing laboratory test, thus has been through a development cycle. The stepper motor direct drive valve approach has essentially been developed in a rotary application which is in production, thus demonstrating the basic principals; also, the two stage valve has been fabricated and tested by NASA's Marshall Space Flight Center. The development effort required would be in adapting the concept to drive a spool sleeve assembly for a linear actuator and the development of the redundant drive circuitry. The stepper motor approach is considered a larger development task.

b. Program Costs Considerations

From a program standpoint, both time and money, the actuation system including control electronics should be designed, developed, and tested as an entity. As such the system specified can be designed, developed and tested concurrent with, and independent of, the flight control computer design. In this manner the development cycle is compatible with a later 1975 flight test schedule. Treating the actuation system as an entity is advantageous in that a clear division of performance requirements, design responsibilities, and testing can be specified for the computer system, the control law development, and the actuation system.

c. Cost Minimization

Utilization of existing actuation hardware was considered for each of the approaches in regard to cost savings. In this investigation, the stepper motor approach was modified to utilize electrical feedback since mechanical feedback would require an entire new actuator. In the directional control system, the existing rudder actuator can and should be used. Due to space limitations, the valve originally was located remote from the actuator. Little could be gained by a new actuator except perhaps integrating the feedback LVDT into the actuator for the direct drive approaches, certainly a desirable feature but insufficient justification for replacement. For the secondary actuator approach, only the secondary actuator would require replacement or modification to improve performance. The existing linkages and mechanical input valve would suffice. For the two direct drive approaches, the mechanical input valve would be replaced with an appropriate valve assembly and a redundant LVDT assembly would be installed to provide position feedback. These modifications can be made with minimal rework to existing hardware.

For the longitudinal control system, the horizontal actuators can be modified to accept mounting of either the PM torque motor or stepper motor valve assemblies. This can be done by fabricating a new actuator body or perhaps by reworking the existing body. This approach saves most of the actuator parts and most of the machining costs; i.e., the piston rod, pistons, end fitting, rod ends, and end caps. The cylinder barrel is of simple straight forward design permitting simple machine operations, primarily boring with a little turning and milling. Fabrication of a new aluminum barrel is estimated to be approximately 10% of a new actuator cost. Another alternative is to block the existing valve and to route hydraulic lines off the actuator to a remote valve package. Hard lines can be used since it is a fixed body actuator installation. Short hard lines have minimal effect upon stiffness and dynamic performance. Remote location allows greater freedom in valve package design which could possibly reduce the design and development cost. In either approach the LVDT position feedback could be mounted external to the actuator. For the secondary actuator approach, the horizontal actuator and control rod linkages can be utilized. The valve may have to be replaced; flow limiting does occur within the specified passband although it is not severe. The final determination is dependent upon the total flight control system design, control laws mechanized, etc.

In the lateral control system, the aileron actuators can be modified to accept either of the direct drive valve assemblies. This can be accomplished by fabricating a new actuator body or possibly by modifying the existing body. This approach utilizes the existing pistons, rods, end caps, and rod ends, etc., a major portion of the actuator, thereby minimizing cost. The LVDT position feedback sensor would be mounted external to the actuator. Another alternative would be to replace the existing valve with a porting block and route cylinder lines off the actuator to a remote valve package. Unfortunately, the F-8 installation design requires a moving body actuator which results in some difficulty in supplying hydraulic fluid to the cylinders. The simplest method, analogous to the existing F-8 installation, is to use hydraulic hoses. Hoses will reduce actuator stiffness by approximately 20%. This is estimated to reduce the total stiffness, including the structure, by 8% and the resonant frequency by 4%. Should these degradations be acceptable, valve packages could be located remote simplifying the design and development effort. For the secondary actuator approach the existing actuators, push rod, and linkages, could be used. The modification required would be replacement or modification of the existing secondary actuators.

4.3 PRELIMINARY DESIGN

The factors considered in the preliminary design were sizing, packaging, and installation.

4.3.1 ACTUATOR SIZING

The design of primary powered controls involves the design and integration of a number of elements into a system. The actuator element must function compatibly within the system. Once control surface size, mass, kinematics, aerodynamic hinge moments, etc., have been established actuators can be sized. The actuators must produce a force which exceeds the maximum aerodynamic hinge moment by an amount sufficient for dynamics. The actuators must also provide adequate stiffness to prevent flutter. The larger of these two requirements establishes actuator size. Establishment of maximum aerodynamic hinge moments is a study of the aerodynamics of the particular vehicle, surface, etc., and is beyond the scope of this study effort. Consequently, it is assumed the existing actuators are sized satisfactorily. Figure 4.3-1 presents force, rate, and stroke data on the existing F-8C actuators.

The frequency at which velocity limiting will occur due to the valve actuator design is given by:

$$\omega = \frac{\dot{X}_{MAX}}{A}$$

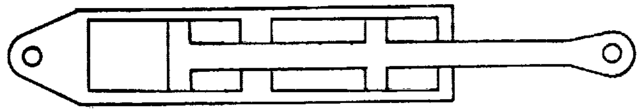
The rate limiting frequency for the existing F-8C actuators with resonably large inputs (i.e., 20%) are:

Rate Limit Frequencies

Ailerons	$\omega = \dfrac{9.0}{\frac{1}{2}(8.52) \times 20\%}$	$= 10.5$ rad/s
Horizontal	$\omega = \dfrac{4.25}{\frac{1}{2}(6.58) \times 20\%}$	$= 6.5$ rad/s
Rudder	$\omega = \dfrac{9.35}{1.6 \times 20\%}$	$= 29.2$ rad/s

Rate limiting does occur within the passband specified for the horizontal actuation system. This limitation can be raised by increasing maximum valve flow rate providing the hydraulic supply is available.

AILERONS



	EXTEND				RETRACT			
	AREA		FORCE		AREA		FORCE	
PC-1	1821 mm^2	(2.823 in^2)	37.7 kN	(8,469 lbs)	1514 mm^2	(2.347 in^2)	31.3 kN	(7,401 lbs)
PC-2	1902 mm^2	(2.948 in^2)	39.3 kN	(8,844 lbs)	1568 mm^2	(2.430 in^2)	32.4 kN	(7,290 lbs)
TOTAL	3723 mm^2	(5.771 in^2)	77.0 kN	(17,313 lbs)	3082 mm^2	(4.777 in^2)	63.7 kN	(14,331 lbs)

Max Rate = 228.6 mm/s (9.0 in/sec)

Stroke = 216.4 mm (8.52 in/sec)

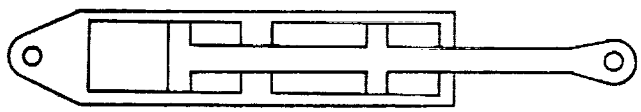
Output Arm = 188 mm (7.4 in.)

Part No. = CV15-901075

Loop Gain = 25 sec^{-1}

Max Flow = 850 cm^3/s (13.5 gpm)

HORIZONTAL



	EXTEND				RETRACT			
	AREA		FORCE		AREA		FORCE	
PC-1	4561 mm^2	(7.07 in^2)	94.3 kN	(21,210 lbs)	3851 mm^2	(5.97 in^2)	79.7 kN	(17,910 lbs)
PC-2	4251 mm^2	(6.59 in^2)	87.9 kN	(19,770 lbs)	3613 mm^2	(5.60 in^2)	74.7 kN	(16,800 lbs)
TOTAL	8813 mm^2	(13.66 in^2)	182.3 kN	(40,980 lbs)	7465 mm^2	(11.57 in^2)	154.4 kN	(34,710 lbs)

Max Rate = 108 mm/s (4.25 in/sec)

Stroke = 143.5 mm (5.65 in.)

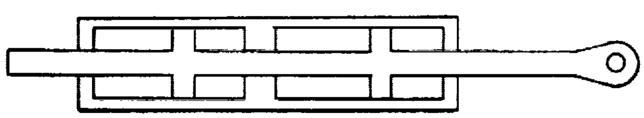
Output Arm = 247.7 mm (9.75 in.)

Part No. = CV15-601051

Loop Gain = 20 sec^{-1}

Max Flow = 1000 cm^3/s (15.9 gpm)

RUDDER



	EXTEND				RETRACT			
	AREA		FORCE		AREA		FORCE	
PC-1	1142 mm^2	(1.77 in^2)	23.6 kN	(5,310 lbs)	1142 mm^2	(1.77 in^2)	23.6 kN	(5,310 lbs)
PC-2	1142 mm^2	(1.77 in^2)	23.6 kN	(5,310 lbs)	1142 mm^2	(1.77 in^2)	23.6 kN	(5,310 lbs)
TOTAL	2284 mm^2	(3.45 in^2)	47.2 kN	(10,620 lbs)	2284 mm^2	(3.54 in^2)	47.2 kN	(10,620 lbs)

Max Rate = 237.5 mm/s (9.35 in/sec)

Stroke = $\pm$40.6 mm ($\pm$1.6 in)

Output Arm = 97.3 mm (3.83 in)

Part No. = CV15-151039-3

Loop Gain = 25 sec^{-1}

Max Flow = 500 cm^3/s (8.6 gpm)

FIGURE 4.3-1 F-8 PRIMARY ACTUATORS

4.3.2 PACKAGING AND INSTALLATION

4.3.2.1 Introduction. Preliminary design effort was conducted for the purpose of identifying any problem or suspected problem areas related to the packaging, sizing, or installation of the conceptual systems which would preclude their development for utilization in the Phase II aircraft. This effort was limited to the PM torque motor direct drive and the stepper motor direct drive systems as the force summed secondary actuator system concept has previously been installed and evaluated in the Phase II aircraft.

4.3.2.2 Package Sizes. The package envelopes for the items of the two conceptual systems are not restrictive at this stage in the program. That is, the details of the package configuration can be varied, within limits, to be compatible with the installation space. To bring the form factors into perspective, the following package sizes were established:

Actuators: Approximately the same as presently required

Stepper Motor	57.2 mm Dia x 152 mm (2¼" Dia x 6")
PM Torque Motor	76 mm x 76 mm x 114 mm (3" x 3" x 4.5")
Valve	38 mm x 76 mm x 127 mm (1.5" x 3" x 5")
Electronics Package	76 mm x 127 mm x 152 mm (3" x 5" x 6")
Dual Stepper Motor Valve Assy	102 mm x 127 mm x 203 mm (4" x 5" x 8")
Dual PM Torque Motor Valve Assy	76 mm x 114 mm x 254 mm (3" x 4.5" x 10")

4.3.2.3 System Baseline Configuration. To establish an installation baseline, the following direct drive (stepper or torque motor) system installation is defined:

a. The electro-hydraulic valve assembly is installed on the power actuator.

b. The power actuator configuration has the Linear Variable Differential Transformer (LVDT) incorporated into the piston rod.

c. The electronics package is mounted in the area immediately adjacent to the actuator/valve assembly.

The baseline configuration minimizes the plumbing between the valves and actuators thereby maximizing actuator stiffness, eliminates the need for external mounting provisions and offers maximum physical protection for the LVDT, and minimizes lead length between the electronic package, LVDT's, and valves.

An aircraft configured with either of the direct drive system concepts as compared to the present day conventional designs, reduces the amount of effort required for installation, rigging, and maintenance. This is due principally to the fact that all mechanical alignments are contained in a single package.

The baseline configuration is of course the ideal. Installation of the actuation system on the Phase II aircraft is dictated by available space, costs, time, and will require alternates to this baseline configuration.

4.3.2.4 Alternatives. The design of all three systems is very versatile and can be varied to match the particular aircraft being considered. The installation shown in the following paragraphs was selected based on the Contractor's judgment of the important program considerations. The design and installation can be changed to be compatible with other program considerations if judged more important. Alternates considered are the remote installation of the electro-hydraulic valve assemblies and installation of the LVDT's external to the actuators.

4.3.2.5 Installation. The installation space, provisions, and locations were determined by reviewing F-8 aircraft handbooks, F-8 drawings, photographs of the Phase II F-8C aircraft, visual examination of the aircraft, and preliminary drawing layouts and sketches. The installation recommended for the Phase II F-8C aircraft makes maximum use of the existing components and assemblies. Power actuator attachments to the aircraft structure and surface linkages are unchanged.

The stepper motor direct drive concept has been evaluated with respect to installation with an electrical feedback in place of the mechanical feedback which was utilized in the comparative analysis. This was done to minimize installation effort on the F-8 aircraft.

The electronic packages would be designed to accept inputs from both the digital computer and the back-up control system.

Location of the five primary power actuators and their respective electronic packages and electro-hydraulic valve assemblies is depicted in Figure 4.3-2.

4.3.2.6 Rudder Actuation System. Figure 4.3-3 depicts the recommended rudder actuation system installation. The electro-hydraulic valve assembly is located remote from the power actuator assembly in an installation similar to that of the existing aircraft configuration. In the directional control system, the existing rudder actuator can and should be used. Due to space limitations, the valve originally was located remote from the actuator. Little could be gained by a new rudder actuator except perhaps integrating the feedback LVDT into the actuator which appears unjustified at this stage of the program. For the two direct drive approaches, the existing mechanical input valve would be replaced with an appropriate valve assembly. The LVDT assembly would be installed to provide position feedback.

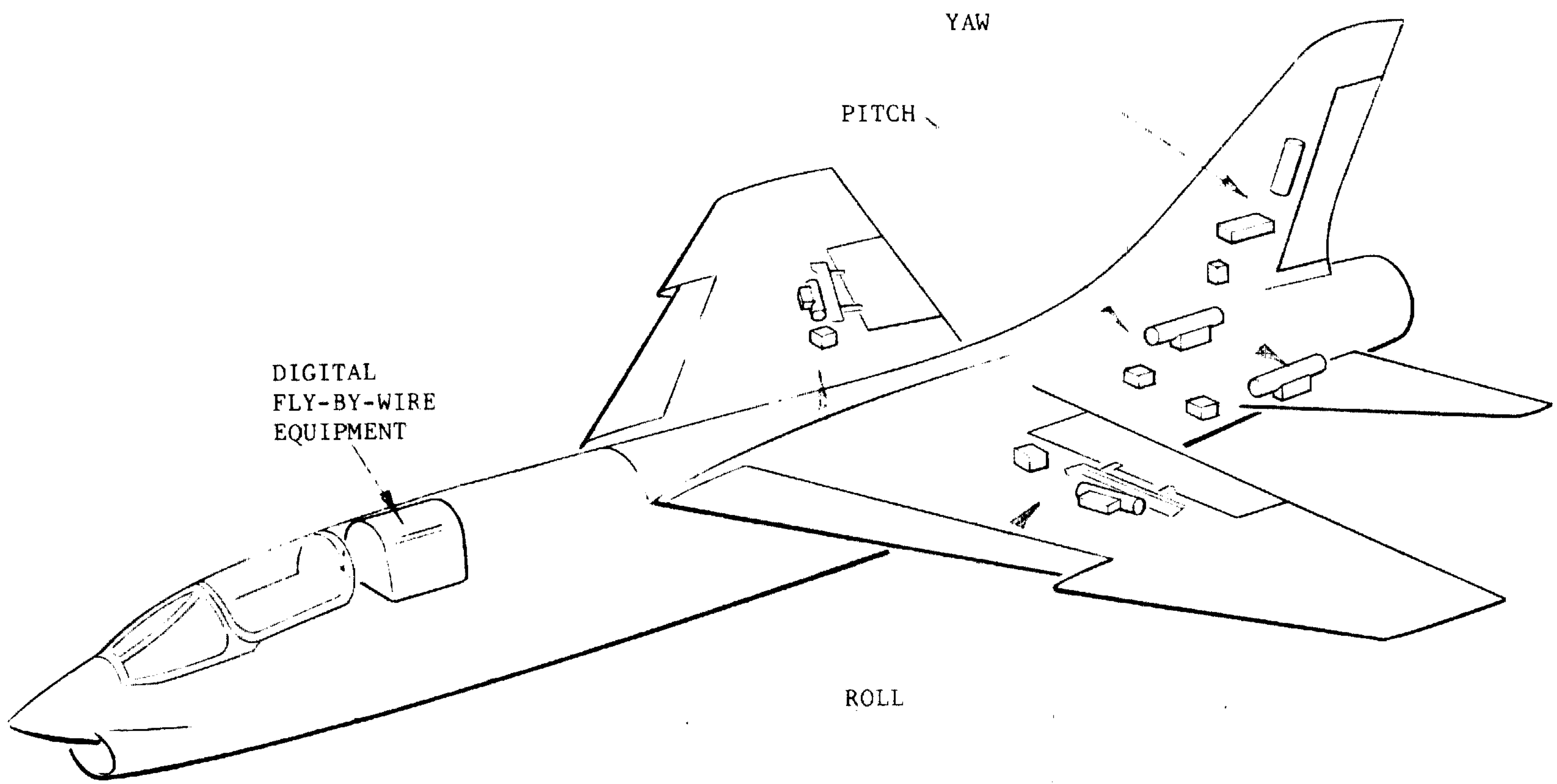


FIGURE 4.3-2 ACTUATION SYSTEM COMPONENT LOCATION

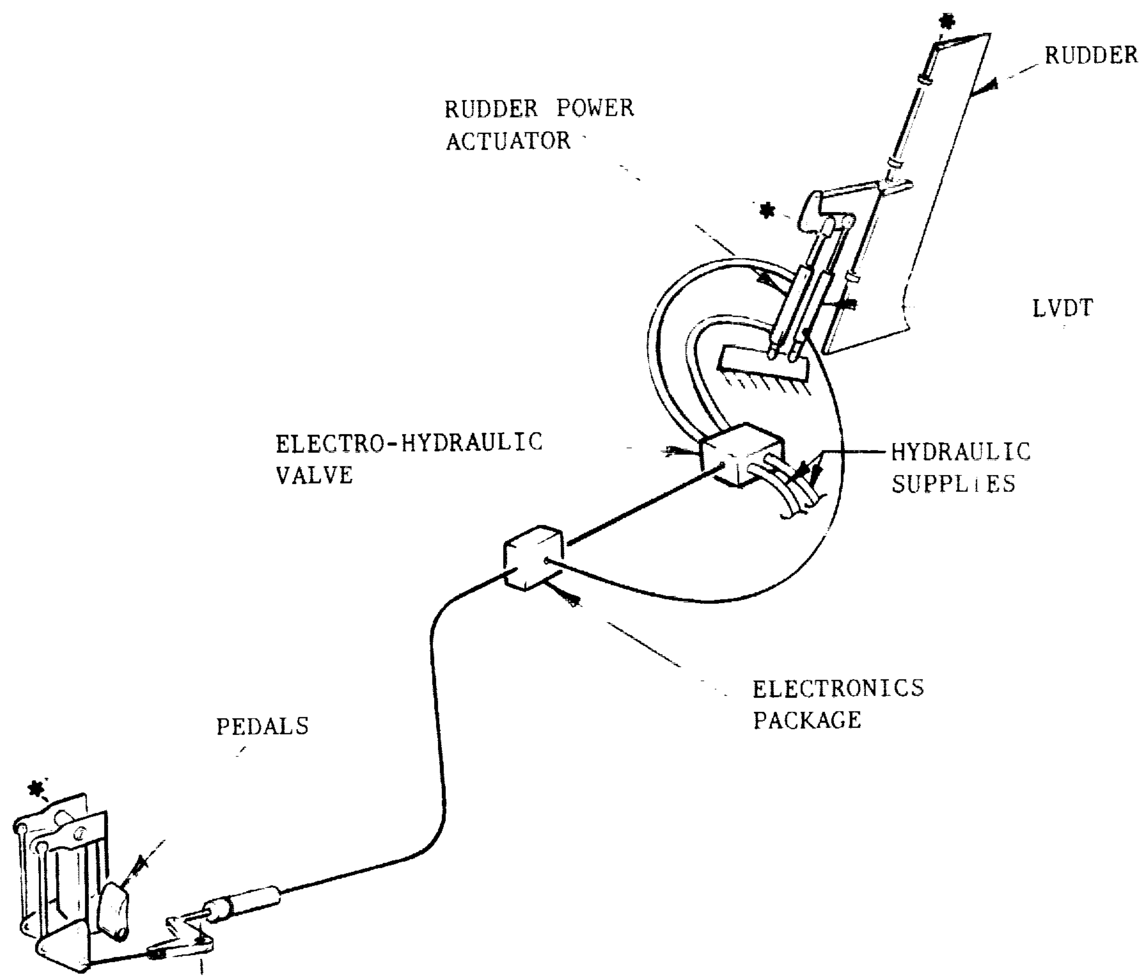


FIGURE 4.3-3 RUDDER ACTUATION SYSTEM

The LVDT is attached to the actuator rod end assembly at the existing feedback rod attachment point. The line routing to the electro-hydraulic valve assembly and from the electro-hydraulic valve assembly to the power actuator is similar to the existing configuration.

The electronics package is installed in the vertical stabilizer forward of the valve assembly in the area where the present secondary actuator is installed.

The above recommended installation uses the Phase II aircraft existing rudder power actuator, thus eliminating the need for design and development of a new actuator. These modifications can be made with very minimal rework to the existing hardware.

4.3.2.7 Horizontal Stabilizer Actuation System. Figure 4.3-4 depicts the recommended horizontal stabilizer actuation system installation. The installation depicted has the electro-hydraulic valve assembly attached to a modified power actuator. The electronics package is installed in the area immediately below the power actuator assembly. The horizontal stabilizer power actuators can be modified to accept mounting of either the PM torque motor or stepper motor valve assemblies. This can be accomplished by fabricating a new actuator body. This approach saves the piston rod, pistons, end fittings, rod ends, and end caps, which amounts to approximately 90% of the fabrication costs of an actuator. As noted in paragraph 4.3.1, the existing power valve/actuator provides marginally acceptable dynamic response and should be redesigned to meet baseline performance requirements.

An alternative to the recommended horizontal stabilizer actuation system installation is to block the existing valve and to route hydraulic lines from the actuator to a remote valve package. Hard lines can be used since the actuator is a fixed body installation. Short hard lines have minimal affect upon stiffness and dynamic performance.

Remote location allows greater freedom in valve package design which could possibly reduce the design and development cost. In each approach the LVDT position feedback is mounted external to the actuator.

4.3.2.8 Aileron Actuation System. Figure 4.3-5 depicts the recommended aileron actuation system installation. The installation depicted has the electro-hydraulic valve assembly attached to a modified power actuator. The electronics package is installed in the area adjacent to the present secondary actuator assembly. The aileron power actuators can be modifed to accept mounting of either the PM torque motor or stepper motor valve assemblies. This can be accomplished by fabricating a new actuator body. This approach saves the piston rod, pistons, end fittings, rod ends, and end caps, which amounts to approximately 90% of the fabrication costs of an actuator. The LVDT position feedback sensor would be mounted external to the actuator. An alternative to the recommended aileron actuation system installation would be to replace the existing valve with a porting block and route hydraulic lines from the actuator to a remote valve package.

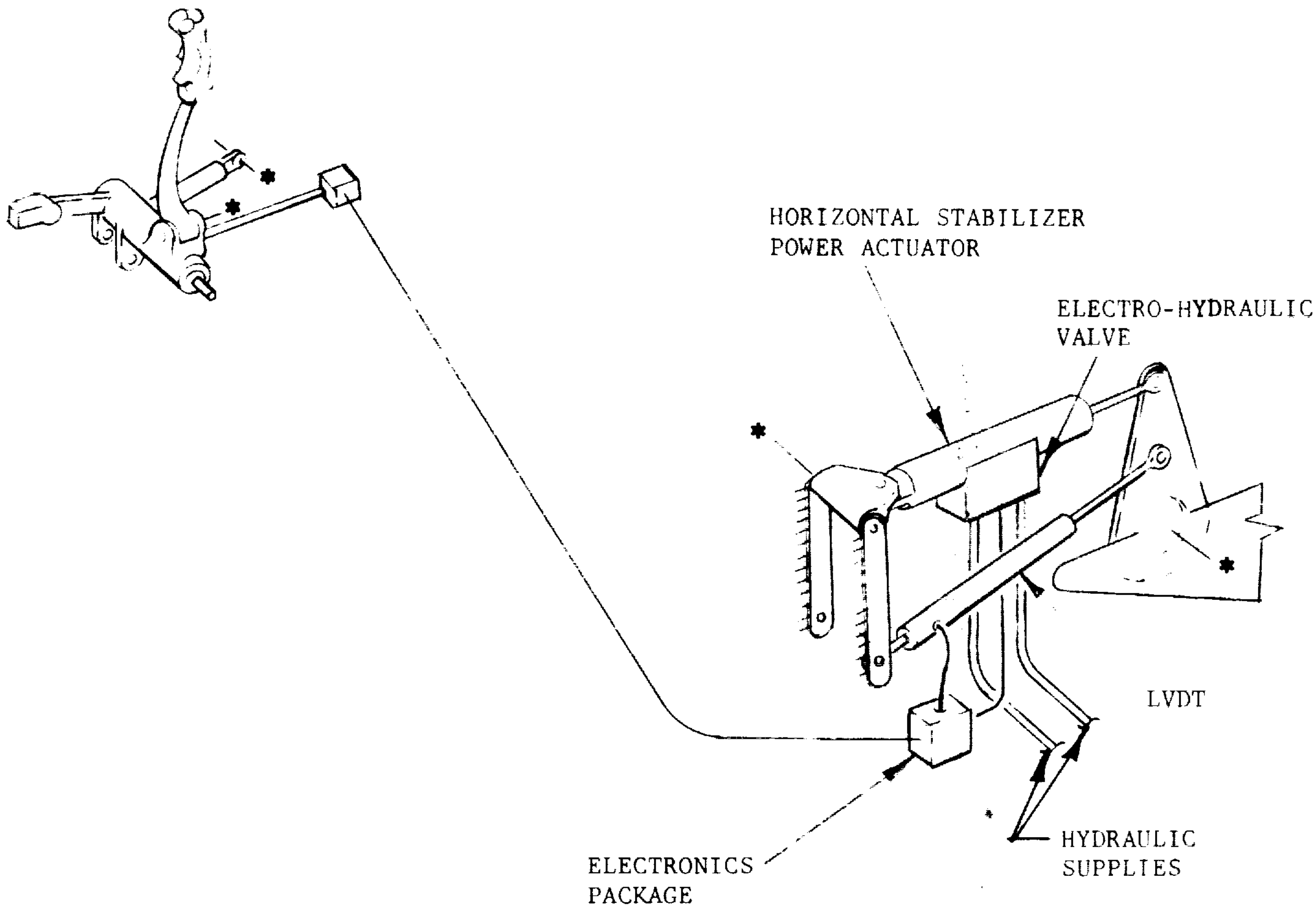


FIGURE 4.3-4 HORIZONTAL STABILIZER ACTUATION SYSTEM

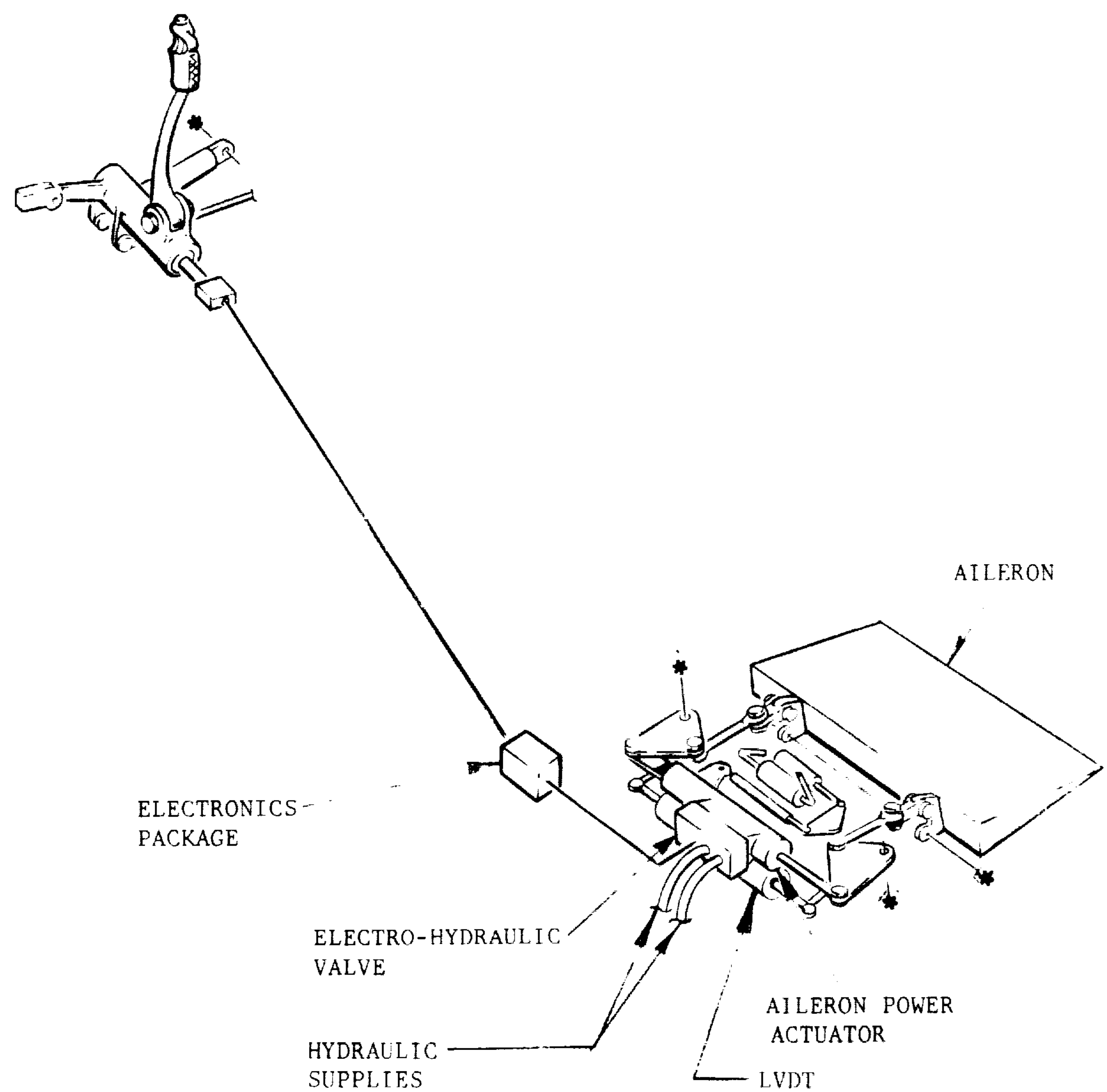


FIGURE 4.3-5 AILERON ACTUATION SYSTEM

The F-8 installation design requires a moving body actuator which results in some difficulty in supplying hydraulic fluid to the cylinders. The simplest method, analogous to the existing F-8 installation, is to use hydraulic hoses. Hoses will reduce actuator stiffness by approximately 20%. This is estimated to reduce the total stiffness, including the structure, by 8% and the resonant frequency by 4%. Should these degradations be acceptable, valve packages could be located remote simplifying the design and development effort.

4.3.2.9 Conclusion. The Contractor does not visualize any problems in designing, developing, and installing any one of the three FBW flight control systems in the Phase II F-8 aircraft. Although the electro-hydraulic valves will differ in flow rates, all three types required will be basically the same. The motor assemblies of the same type would be identical. The actuator body design is anticipated to be similar to the existing actuators.

The LVDT's would all be similar in design. The electronics packages would be identical for each system.

4.4 CONFIGURATION SELECTION

The preceding paragraphs presented a trade-off of the factors involved in the application of the three configurations, selected from the comparative analysis, to the proposed phase II aircraft. Paragraph 4.3 undertook a preliminary design to identify any problem areas which would preclude development of any of the three configurations for use on the phase II aircraft.

The preliminary design did not uncover any problem area which would preclude using any of the three configurations on the phase II aircraft. All three designs are sufficiently flexible and offer many trade-offs in the actual design for the phase II aircraft. Program considerations of time, cost, and performance needed for the control laws being investigated, will determine the ultimate actuation system design. The three configurations selected for preliminary design would all be compatible with concurrent and separate development efforts on the computer system and the control law mechanizations.

The trade-off indicated the PM torque motor direct drive configuration will best satisfy the requirements of the NASA digital fly-by-wire program. It is recommended that this configuration be developed for the initial flight program utilizing the multi-channel digital fly-by-wire system. The PM torque motor direct drive configuration was selected primarily on basis of performance, reliability, simplicity, and compatibility with future digital fly-by-wire requirements. Table 4.4-1 is an accumulation of comparative data on the three design concepts. The paragraph numbers in Table 4.4-1 refer to paragraphs in this report where a more detailed discussion of the parameters can be found. In summary, the PM torque motor direct drive approach was selected because of the following advantages:

1. High Performance
2. High Reliability
3. Low Maintenance
4. Rugged
5. Simple Direct
6. Compatible with Advanced Actuator Design
7. High Power Efficiency
8. Low Weight
9. Low Sampling Rate Req. for Digital Loop Closure
10. Compatible with Advanced Hydraulic Systems

TABLE 4.4-1 ACCUMULATION OF COMPARATIVE DATA

PARAMETER \ CONFIGURATION	PM TORQUE MOTOR DIRECT DRIVE	SECONDARY ACTUATOR	STEPPER MOTOR DIRECT DRIVE
Performance Paragraph 4.2.1 3.2.2 3.3.1 3.3.2 3.4.5	Highest. Offers most potential for future applications. Power actuator is the limiting component.	Adequate for F-8. Limited for active control law applications. Friction is important design consideration.	Adequate for F-8. Limited for active control applications.
Failsafe Reliability Paragraph 4.2.2 3.2.3 3.3.1 3.3.2 3.4.1	$.56 \times 10^{-6}$ FPFH	$.57 \times 10^{-6}$ FPFH	$.62 \times 10^{-6}$ FPFH
	The power actuator is the predominant factor. Selection of three or four actuation system channels dependant upon flight control system design, not upon actuation system reliability.		
Maintenance Reliability Paragraph 4.2.2 3.2.3 3.3.1 3.3.2 3.4.2	930×10^{-6} 4 CH 860×18^{-6} 3 CH	1240×10^{-6} 4 CH 1090×10^{-6} 3 CH	1120×10^{-6} 4 CH 1010×10^{-6} 3 CH
Total System Fail-safe Reliability Paragraph 4.2.2 3.2.3	1.96×10^{-6} FPFH	1.96×10^{-6} FPFH	2.01×10^{-6} FPFH
	The hydraulic power system and power actuators are the predominant factors.		
Number of Channels Paragraph 4.2.3.1	Three recommended - fourth channel decreases maint. reliability by 7.5% and has minor effect on failsafe reliability.	Three recommended - fourth channel decreases maint. reliability by 12% and has minor effect on failsafe reliability.	Three recommended - fourth channel decreases maint. reliability by 10% and has minor effect on failsafe reliability.
Valve Driver Paragraph 4.2.3.2	Best. Offers most potential for future applications.	Adequate for F-8. Limited for future high performance applications.	Adequate for F-8. Limited for future high performance applications.

TABLE 4.4-1 ACCUMULATION OF COMPARATIVE DATA (cont'd)

CONFIGURATION / PARAMETER	PM TORQUE MOTOR DIRECT DRIVE	SECONDARY ACTUATOR	STEPPER MOTOR DIRECT DRIVE
Interface Electronics Paragraph 4.2.3.3 3.3.3	Adequate	Adequate	Adequate
	Compatible with either 3 or 4 channel computer mechanization. Can accommodate backup analog channels.	Compatible with either 3 or 4 channel computer mechanization. Can accommodate backup analog channels.	
Loop Closure Paragraph 4.2.3.4 3.3.3	Recommend loop closure outside computer. Lighter weight, less volume. Nothing precludes closing loop through computer.	Recommend loop closure outside computer. Lighter weight, less volume. Nothing precludes closing loop through computer.	Recommend loop closure outside computer. Lighter weight, less volume. Nothing precludes closing loop through computer.
Monitor Paragraph 4.2.3.5	Not required by actuation system. Incorporation for total system considerations simplest.	Not required by actuation system. Incorporation for total system considerations, more complicated than P/M torque motor.	Required by actuation system. Incorporation more complicated than P/M torque motor.
Power System Impact Paragraph 4.2.4 3.3.1	Lowest Quiescent power consumed 270 watts for 3 channels, 275 watts for 4 channels.	Highest Quiescent power consumed 4120 for 3 channels, 5400 watts for 4 channels.	Low Quiescent power consumed 310 watts for 3 or 4 channels.
Economic Factors Paragraph 4.2.5	Most economical	Higher cost than the PM torque motor approach.	Higher cost than the PM torque motor approach.
	All three configurations are flexible in design and offer many trade-offs in economic factors.	All three configurations are flexible in design and offer many trade-offs in economic factors.	All three configurations are flexible in design and offer many trade-offs in economic factors.
Sizing Paragraph 4.3.1	Adequate	Adequate	Adequate
	Existing power actuators are satisfactory. The horizontal is marginal and would require replacement if control law investigation required higher performance.	Existing power actuators are satisfactory. The horizontal is marginal and would require replacement if control law investigation required higher performance.	Existing power actuators are satisfactory. The horizontal is marginal and would require replacement if control law investigation required higher performance.
Installation Paragraph 4.3.2	Adequate	Adequate	Adequate
	All three configurations are compatible with installation of the F-8 aircraft. The installation offers many trade-offs in economics and ease of installation.	All three configurations are compatible with installation of the F-8 aircraft. The installation offers many trade-offs in economics and ease of installation.	All three configurations are compatible with installation of the F-8 aircraft. The installation offers many trade-offs in economics and ease of installation.
Packaging Paragraph 4.3.2	Adequate	Adequate	Adequate
	All three configurations are sufficiently flexible in pack. design.	All three configurations are sufficiently flexible in pack. design.	All three configurations are sufficiently flexible in pack. design.

4.5 MATHEMATICAL MODEL

Typical torque motor characteristics are shown in Figure 4.5-1. The device operates fairly linearily over the entire envelope and is very linear in the null region. The torque motor is adequately represented by the following equation:

$$F = K_T \Delta i - K_X x_V$$

where
F = output force
Δi = coil current
x_V = output displacement
K_T = torque motor torque coefficient
K_X = torque motor spring coefficient

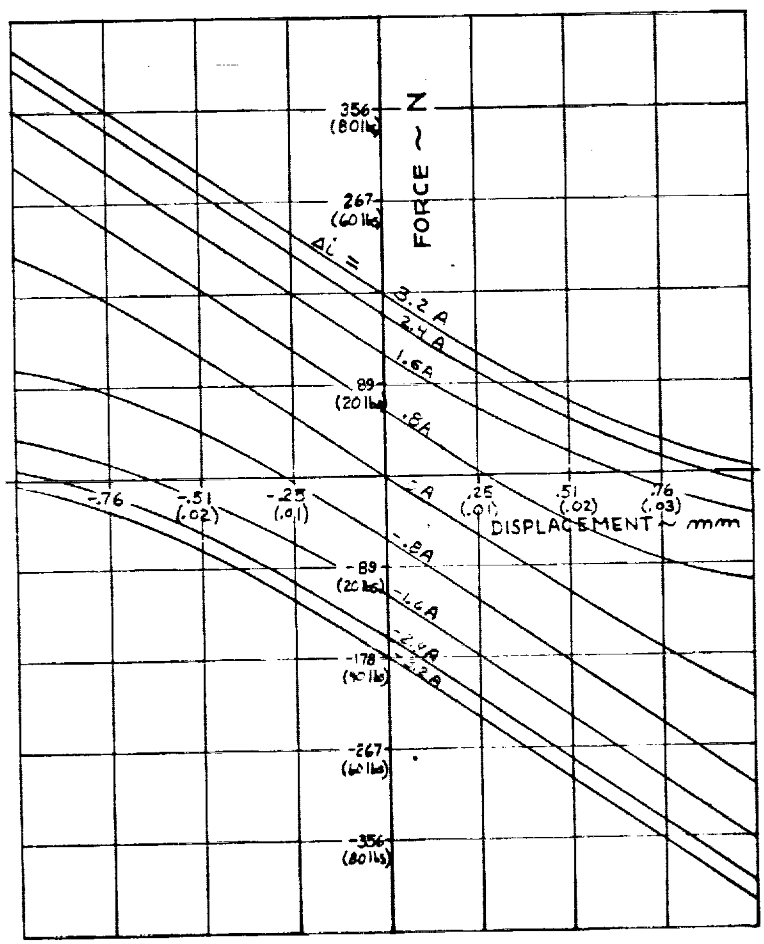


FIGURE 4.5-1 PM TORQUE MOTOR CHARACTERISTICS

Each torque motor has 4 coils. Each coil is driven by an amplifier which is limited in output such that its maximum output current can produce $\frac{1}{4}$ of the saturation flux and consequently $\frac{1}{4}$ the maximum torque output. Each channel drives one coil in each of the two torque motors via individual valve driver amplifiers. A math model for one PM torque motor is depicted in Figure 4.5-2.

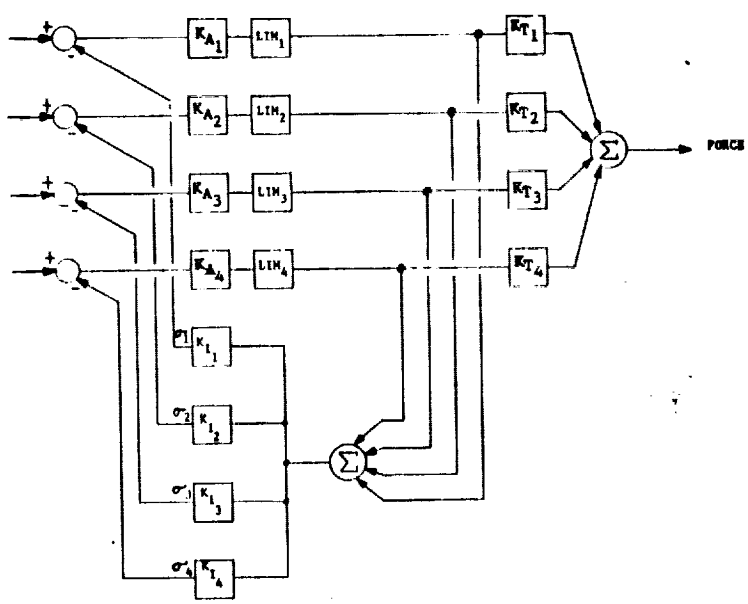


FIGURE 4.5-2 TORQUE MOTOR BLOCK DIAGRAM

The dual tandem valve and actuator are assumed to operate in exact synchronism and can therefore be represented by a single valve-actuator arrangement. Experience proves this simplifying assumption to be practically valid. A linearized description of the valve, actuator, and load is presented in the following equations. The basic torque motor equations are also repeated.

Electrical Circuit Eq.

$$\Delta i = K_A \epsilon$$

where, K_A = Current amplifier gain

ϵ = Servo error

Δi = Torque motor current

Torque Motor Eq.

$$F = K_T i + K_x X_v$$

where, K_T = Torque coefficient

K_x = Spring coefficient

F = Torque motor output

X_v = Valve position

Torque Motor Load Eq.

$$F = \frac{1}{2} (M_v \ddot{X}_v + B_v \dot{X}_v)$$

where, M_v = Mass of armature + spool

B_v = Damping of armature + spool

Valve Flow Eq.

$$q = K_q X_v - K_c P$$

where, K_q = Valve flow gain

K_c = Valve pressure/flow coefficient

q = Flow

Actuator Flow Eq.

$$q = K_\beta P + C_{tp} P + A_p \dot{x}_p$$

where, K_β = Compressibility coefficient

C_{tp} = Piston leakage coefficient

A_p = Piston area

P = Load pressure

x_p = Piston position

Actuator Load Eq.

$$P A_p = M_L \ddot{x}_p + B_L \dot{x}_p + K_L x_p$$

where, M_L = Load mass

B_L = Load damping

K_L = Load spring

A linearized mathematical model of the PM torque motor direct drive approach is depicted in Figure 4.5-3. Estimated coefficients are presented in Table 4.5-1.

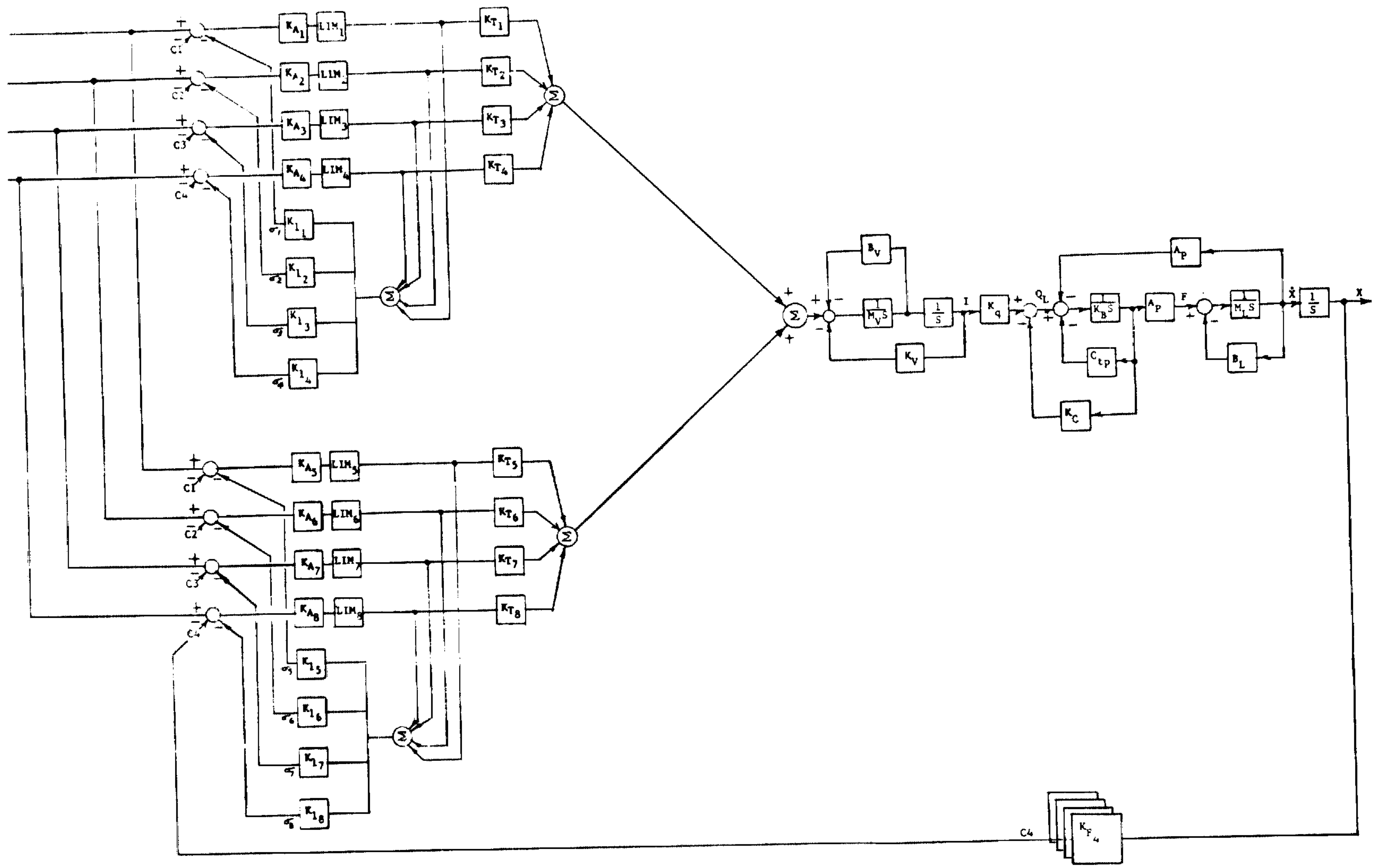


FIGURE 4.5-3 PM TORQUE MOTOR APPROACH BLOCK DIAGRAM

TABLE 4.5-1 PM TORQUE MOTOR MODEL COEFFICIENTS

COEFFICIENT	VALUE	
A	400 A/V	-
K_I	0.25 V/A	-
K_T	55.6 N/m	(12.5 lbs/A)
K_x	.224 MN/m	(1280.0 lbs/in)
LIM	0.8 A	
M_v	.507 N/m	(0.0029 lb-sec^2/in)
B_v	96.3 N•s/m	(0.55 lb-sec/in)
K_v	.448 MN/m	(2560.0 lbs/in)
K_q	.294 m^3/s	(456.0 in^3/sec)
K_β	532 mm^5/lb	(2.24 x 10^{-4} in^5/lb)
C_{tp}	237. mm^5/N•s	(.0001 in^5/lb-sec)
K_c	0	-
A_p	8840 mm^2	(13.7 in^2)
M_L	1.22 kN•s/m	(6.95 lb-sec^2/in)
B_L	4.2 kN•s/m	(24.0 lb-sec/in)
K_L	0 N/m	
K_F	.38 to .51 V/m	(15 to 30 V/in)

The block diagram in Figure 4.5-3 is a rather detailed linear representation (with exception of the limiters) of the PM Torque Motor Approach useful for studying the effects of failures within the actuation system. The major nonlinearities ignored by this representation are valve orifice flow, valve lap characteristics, actuator/load friction and backlash all of which can effect stability and performance. These characteristics are well known and can easily be added to the model if desired, however, the projected use of the model does not justify the added complexity. A simpler model is often useful in studying the overall system operation and the effect of system input failures to the actuation system. The model of Figure 4.5-3 can be simplified by the following assumptions.

(1) Assume the amplifiers, coils, limiters, etc., associated with each channel are identical.

(2) Assume the valve dynamics is far beyond the passband.

(3) Assume a first order representation of the valve/actuator/load.

The simplified model is shown in Figure 4.5-4.

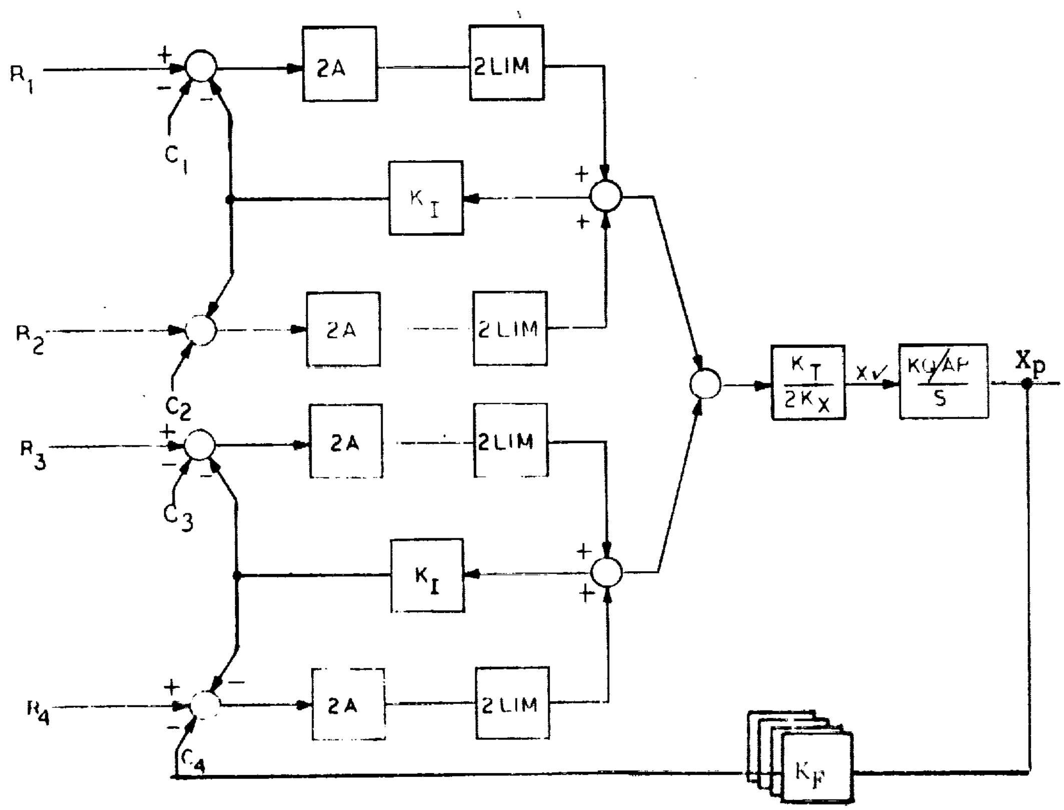


FIGURE 4.5-4 PM TORQUE MOTOR DIRECT DRIVE SIMPLIFIED BLOCK DIAGRAM

SECTION 5

RECOMMENDATIONS

The purpose of the research study described herein was to review and advance the state-of-the-art of flight control actuators and feedback sensors suitable for use in multi-channel digital flight control systems, and to prepare the most appropriate conceptual design of an advanced digital flight control actuator and feedback sensor system which would be suitable for use in Phase II flight tests. One goal of the fly-by-wire experimental program is to demonstrate advanced flight control concepts which may include forms of adaptive or optimal control, active controls technology, or control configured vehicle applications. These advanced applications necessitate that the flight control actuation system has improved performance with appropriate fast response, accuracy, and reliability over a wide range of environmental conditions. The goal of this study was to provide the most appropriate advanced flight control actuation and feedback sensor system design concept which can meet the requirements of a digital fly-by-wire control system and which can be developed on a timely basis for a flight demonstration on the Phase II test aircraft in the 1976 time period. The further intent was that the actuator and feedback sensor design concepts resulting from this study be sufficiently general that they could be applicable for use on future aircraft which might utilize the advanced, redundant, digital flight control concepts demonstrated in the experimental fly-by-wire program.

The recommendations presented here are made in consideration of the goals described above. Some of the recommendations appear elsewhere in the report but are repeated here for convenience.

The Contractor recommends that the PM torque motor direct drive actuation system concept be developed for use on the Phase II aircraft. The system should be installed on all three axis of the Phase II aircraft and should be developed to be installed on the first flight of the all digital system. The actuation system includes the necessary valve driver electronics. Development should include the design, functional tests, air worthiness tests, and simulator tests prior to installation on the Phase II aircraft.

The existing F-8C secondary actuators should be replaced or extensively modified. The secondary actuators currently being used do not meet the goals of performance with approporate fast response, accuracy, and applicability for use on future aircraft which utilize the advanced, redundant, digital flight control concepts being demonstrated in the experimental program. The existing horizontal stabilizer power valve/actuator provides marginally acceptable dynamic response and should be redesigned to meet baseline performance requirements.

The concept of direct electro-mechanical (EM) actuation of the aircraft control surface is recommended for further development. The effort envisioned is specific hardware design, development, and testing. This effort should be conducted and coordinated with the overall FBW program to produce flight-worthy hardware for incorporation into the F-8 test vehicle in the 1977 period. EM technology has progressed to the point where specific design and test data are required to advance the concept.

A study of the entire power generation and distribution systems in aircraft should be conducted. The study should consider: weight, efficiency, reliability, maintainability, single and multi-engine aircraft, high voltage AC and DC, high frequency power, centralized and localized power distribution, very high pressure hydraulics, and integrated actuator packages.

A systems design study should be conducted to establish the requirements for the entire flight control system.

The flight control actuation servo loops should be closed external to the main flight computers. This can be accomplished with less weight, less power consumption, and in a smaller size than closing the loops through the digital computer.

APPENDIX A

STATE OF THE ART SURVEY AND REVIEW

CONCEPT CATEGORIZATION

Figures A-1A, A-1B, and A-1C are flow charts showing three major areas considered in this study. The flow charts are further broken down within each of the major areas to eventually show the applicable concepts contained in this Appendix. The three major areas are; Digital Hydraulic Actuation (Actuators), Redundant Actuation (primary actuator systems), and Components, Systems, and Integrated Actuator Packages applicable to a fly-by-wire concept. An explanation of each flow chart is given in the following paragraphs.

FIGURE A-1A DIGITAL HYDRAULIC ACTUATION

Digital hydraulic actuation concepts can be classified in accordance with Figure A-1A. Basically they fall within two groups, quantized rate or quantized position. Typical quantized rate type concepts found were flow summation i.e., flow from on-off valves are summed to produce total actuation rate - DIGICON, stepper motor driven flow control valves, and various approaches to a digital driving of analog type valves. The quantized position concepts reviewed operate on a digitized volume or digitized step principle. Digitized volume devices move incrementally by injecting a fixed quantity of fluid into the actuator. Digitized stepping devices utilize some form of feedback to close the valve after it has moved the commanded increment. Examples of each of the concepts are presented in this Appendix with concept index numbers listed in Figure A-1A.

FIGURE A-1B REDUNDANT ACTUATION

Considerable effort has been expended over the past decade in the study and development of redundant actuators and actuation systems as attested by the volume of literature available. The evolution began from simple limited authority fail safe parallel or series actuators for electronic flight control commands (SAS) and has progressed through one and two fail operational full authority fly-by-wire systems. The Redundant Actuation concept can be classified in accordance with Figure A-1B. The first level of breakdown is the basic approach to redundancy; active, standby, or some combination of two, hybrid redundancy. Most of the work to date has been in either the active or standby categories. The first level breakdown can be further divided into groups defining the power level to which the redundancy is carried; control power level or output power level. For example, a standby system which utilizes multiple secondary actuators to control the surface actuator would be listed under the control power category. The control power category can be further divided by whether the redundant channels combine at a secondary actuator or at a direct drive valve. As shown in Figure A-1B a fourth and fifth

FIGURE A-1B REDUNDANT ACTUATION (Continued)

level categorization is made in the active redundancy branch depending upon whether the system utilizes force, rate, or position summation and further by whether the secondary actuator is hydraulic or electro-mechanical. Examples of each category can be found in this Appendix from the concepts listed. As indicated by the number of examples, the active redundant, secondary, force summed, hydraulic actuator and the standby, secondary actuator approaches have been selected in the majority of the past development efforts.

FIGURE A-1C COMPONENTS, SYSTEMS, AND INTEGRATED ACTUATOR PACKAGES

A representative collection of Components, Systems, and Integrated Actuator Packages (IAPS), found in the literature search and survey which are applicable to digital control systems or primary flight control systems are included in this Appendix. Figure A-1C lists by index number those concepts presented.

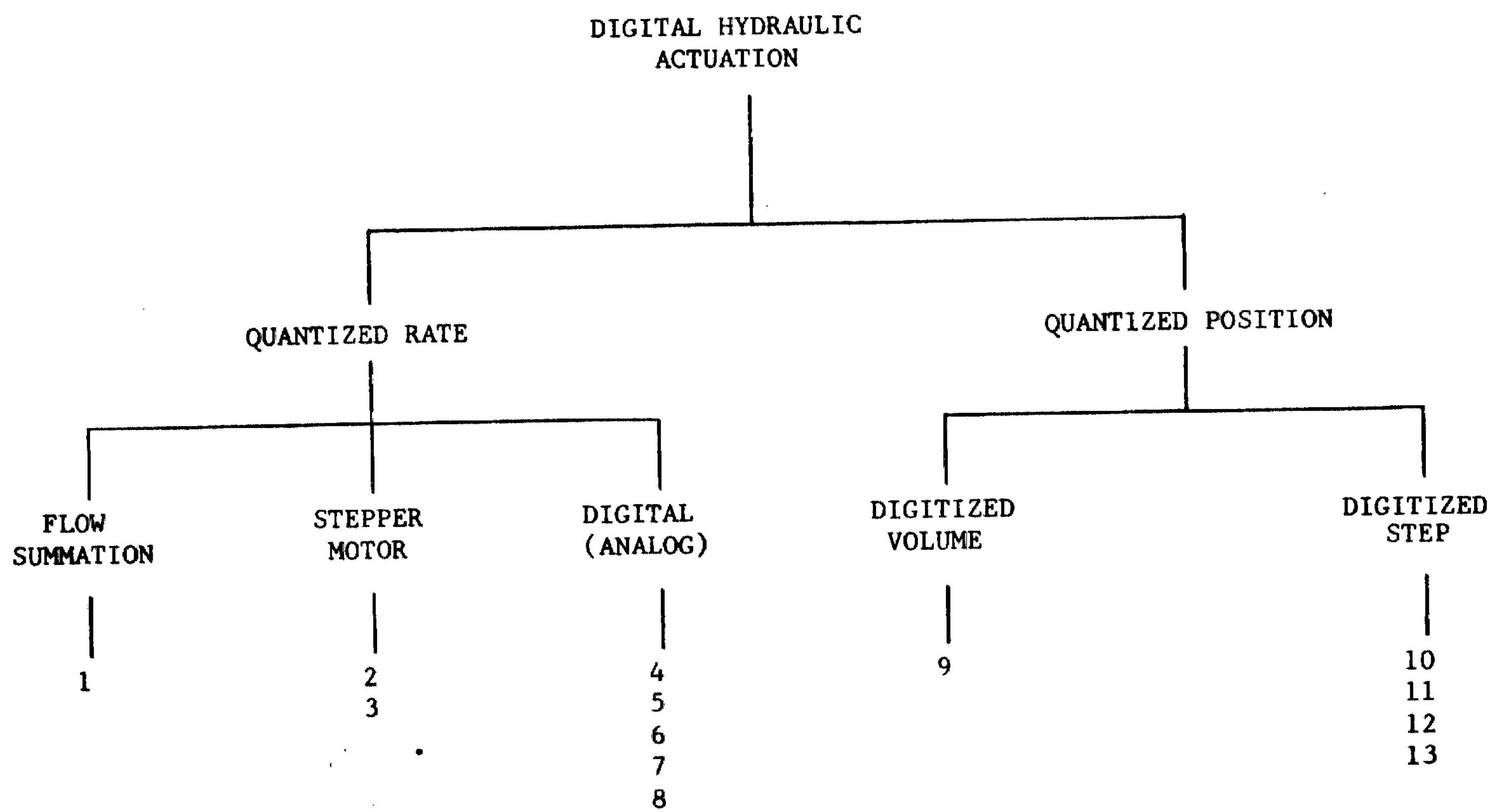


FIGURE A-1A FLOW DIAGRAM-DIGITAL HYDRAULIC ACTUATION

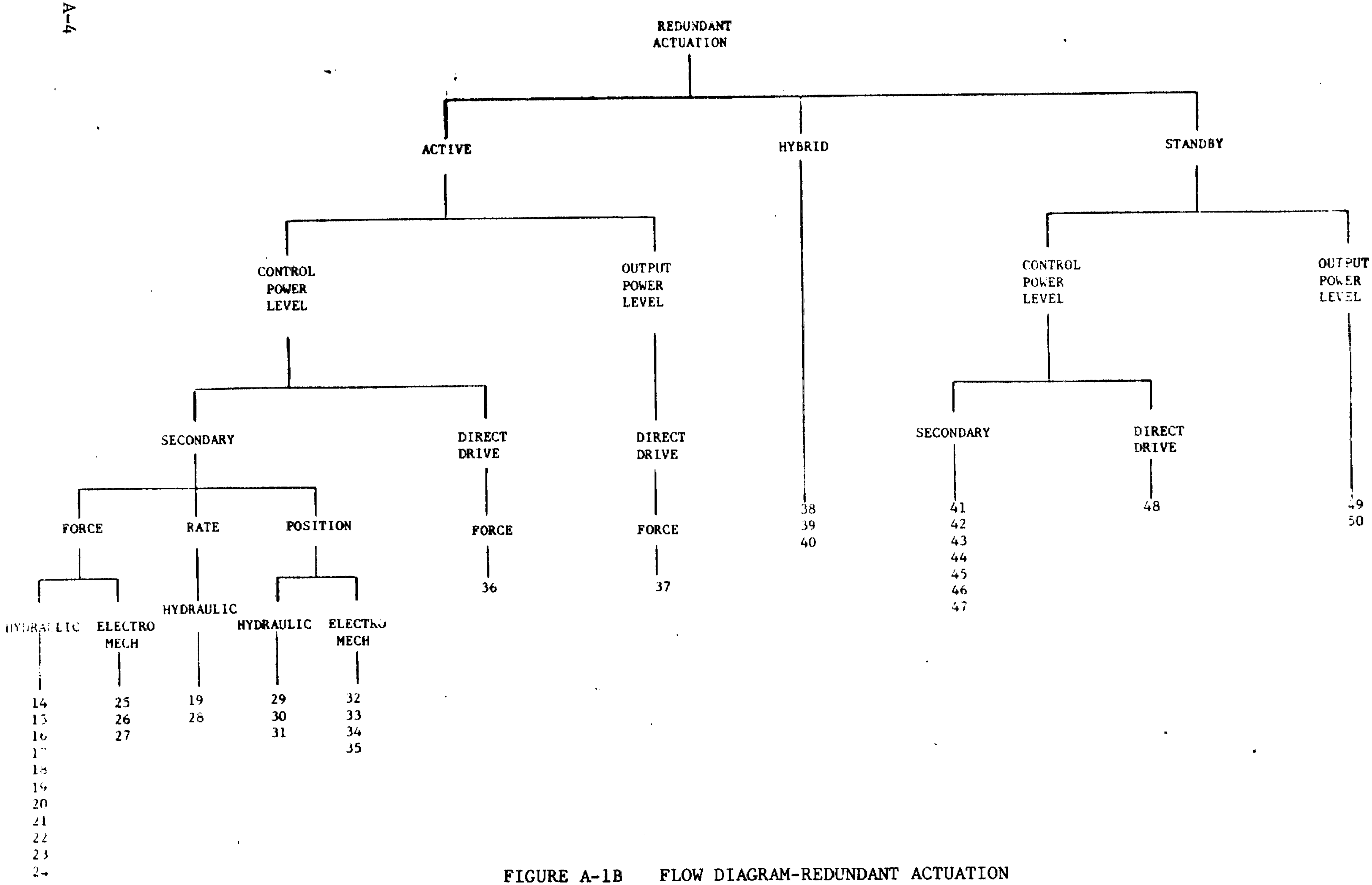


FIGURE A-1B FLOW DIAGRAM-REDUNDANT ACTUATION

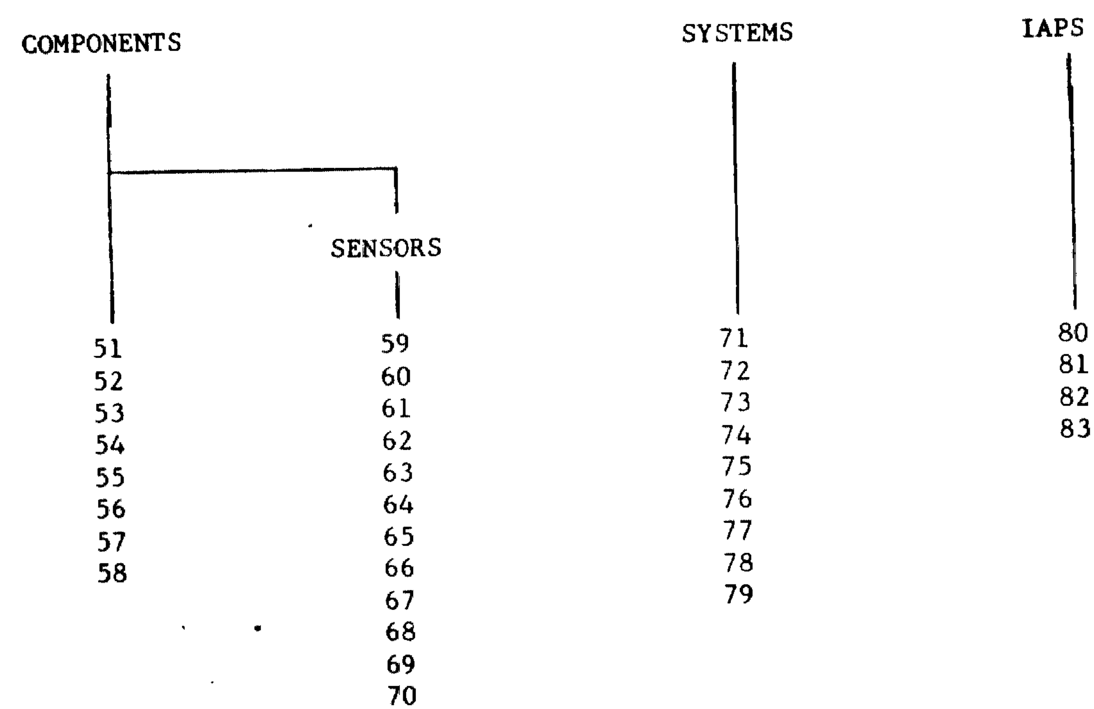


FIGURE A-1C FLOW DIAGRAM-COMPONENTS, SYSTEMS AND IAPS

SUMMARY MATRIX

The concepts presented in this Appendix are summarized, in matrix form, in Figure A-2 which is subdivided into groups; i.e. digital hydraulic actuation, redundant actuation, components, etc. The areas of interest included Concept Classification (direct drive, digital, analog), Identification (association with current programs like F-14, F-15, Space Shuttle, 680J), Redundancy (fail operate, fail safe, fail to null), Control (jet pipe, flapper, clutch, solenoid), Power (hydraulic, pneumatic, fluidic, mechanical), Power Source (aircraft engine, electric motor, battery), Actuators (dual or triple tandem and dual, triple and quadruplex parallel), Feedback Sensors (LVDT, RVDT, CAMS), Monitoring (pressure, hydraulic position, electrical), Status (conceptual, experimental, breadboard, prototype, production), and any comments concerning the concept itself.

CLASS	VALVE	DIGITAL POWER	DIGITAL-SERVO	DIGITAL/SERVO
IDENTIFICATION	PROCESS SYSTEMS, INC. DIGITAL CONTROL VALVE	PULSE MOTOR ICON CAMBRIDGE, MASS. INDUSTRIAL	HLM INCORPORATED PULSE MOTOR	PARALLEL BINARY SYS. STUDY-HYDR. RES & MFG. CO.
DESIGN CONCEPT	1-107	2-161	3-166	4-049
REDUNDANCY	N/A	FORCE SUMMING	N/A	N/A
CONTROL	ON-OFF DIGITAL WEIGHTED OFF to 255	STEPPING MOTOR & VALVE	HYDRAULIC HELIX COUPLED PILOT SPOOL	MULTI COIL FLAPPER VALVE
POWER	PNEUMATIC HYDRAULIC	HYDR	HYDR	HHDR
POWER SOURCE	N/A	AIRCRAFT SUPPLIES	N/A	N/A
ACTUATOR DESCRIPTION	N/A	HYDRAULIC MOTOR	N/A	SINGLE
FEEDBACK	OPEN LOOP	MECHANICAL	POSITION	MECHANICAL
MONITORING	N/A	NONE	N/A	N/A
STATUS	INDUSTRIAL APPLICATIONS	PRODUCTION	EXPERIMENTAL	EXPERIMENTAL-CONSTRUCTED & TESTED BREADBD.
COMMENT	NOT ACFT TESTED NEEDS DEVELOPMENT FOR DIR. DGTL. CONT.			

DIGITAL HYDRAULIC ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	DIGITAL/SERVO	DIGITAL/ANALOG SERVO	ANALOG/SERVO	DIGITAL/SERVO
IDENTIFICATION	BINARY SERVOVALVE SHPBD MSL LCHR-HYDR RES & MFG CO.	HYBRID DIGITAL SERVO AUTONETICS DIV	SERVO VALVE - FLAPPER TOKYO INSTITUTE OF TECH	DIGITAL ACTUATOR
DESIGN CONCEPT	5-082	6-061	7-168	8-173
REDUNDANCY	N/A	ACTIVE	N/A	N/A
CONTROL	DIGITAL TORQUER	DIGITAL SERVO VALVE	PULSE WIDTH MODULATION CONTROLLED FLAPPER	PULSE WAVE MODULATED FLAPPER
POWER	HYDR.	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	SINGLE	N/A	N/A	SINGLE
FEEDBACK SENSOR	MECHANICAL	PARALLEL OR INCREMENTAL (ELECTRICAL)	TACHOMETER	ELECTRONIC
MONITORING	N/A	N/A	ELECTRIC	POSITION
STATUS	EXPERIMENTAL-MODELS BUILT & TESTED	BREADBOARD UNIT EVALUATED DEVELOPMENT	EXPERIMENTAL	CONCEPTUAL
COMMENT	20 WORD/SEC INPUT RATE & LESS THAN 1% RESOLUTION SHOWN			

DIGITAL HYDRAULIC ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	DIGITAL/POWER	DIGITAL/POWER	DIGITAL SERVO	DIGITAL/SERVO
IDENTIFICATION	DIGITAL SYS STUDY HYD. RES & MFG. CO.	GEAR CAM DIGITAL SYS. STUDY-HYD. RES & MFG. CO.	STEPPER ACTUATOR D.B.A.- AIR EQUIPMENT	DIGITAL LINEAR SERVO ACTUATOR, MSL & ACFT CADILLAC GAGE CO.
DESIGN CONCEPT	9-049	10-049	11-141	12-073
REDUNDANCY	N/A	N/A	POSITION SUMMING	N/A
CONTROL	POPPET VALVE	SOLENOID VALVES	HYDR CONTROL VALVE	SOLENOID VALVES
POWER	HYDR.	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	SINGLE	SINGLE HYDR.	SINGLE	"Jo" BLOCK
FEEDBACK SENSOR	NONE	MECHANICAL	MECHANICAL	NONE
MONITORING	N/A	N/A	NONE	N/A
STATUS	EXPERIMENTAL-CONSTRU-CTED & TESTED BREADBD.	CONCEPTUAL (NO BREAD-BOARD)	COMMERCIAL MODEL AVAILABLE	PRODUCTION
COMMENT	LOW SLEW RATE			LARGE, NOISEY, SLOW DYNAMIC PROBLEMS

DIGITAL HYDRAULIC ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	DIGITAL/POWER			
IDENTIFICATION	DIGITAL ACTUATOR VICKERS INC.,DIV SPERRY RAND			
DESIGN CONCEPT	13-003			
REDUNDANCY	N/A			
CONTROL	PARALLEL 8 BIT BINARY CODED INPUT			
POWER	HYDR.			
POWER SOURCE	N/A			
ACTUATOR DESCRIPTION	SINGLE			
FEEDBACK SENSOR	NONE			
MONITORING	N/A			
STATUS	DEVELOPMENT			
COMMENT				

DIGITAL HYDRAULIC ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG SERVO	ANALOG/SERVO	
IDENTIFICATION	SECONDARY ACTUATOR GEN ELECTRIC CO. F4 ACFT-680J PROJ	SERVO ACTUATOR MODEL 3549 NATL W.L. W.L. CO.& HONEYWELL	FLY-BY-WIRE SYS. SPERRY RAND/WESTON HYDRAULIC	
DESIGN	14-029	15-106	16-016	
REDUNDANCY	FORCE SUM FO/FO/FS	QUAD FORCE SUM	FORCE SUMMING	
CONTROL	JP VALVE	JET PIPE SERVO VALVE	SERVOVALVE	
POWER	HYDR	HYDRAULIC	HYDR	
POWER SOURCE	N/A	N/A	N/A	
ACTUATOR DESCRIPTION	4 PARALLEL	QUAD. CHANNEL SERVO ACTUATOR	TRIPLE PARALLEL	
FEEDBACK SENSOR	QUAD-LVDT	L.V.D.T	RVDT.	
MONITORING	ELECTRONIC ($\triangle$P)	HYDRO/MECH.	DIFFERENTIAL PRESSURE TRANSDUCERS	
STATUS	FLOWN	EXPERIMENTAL	DEVELOPMENTAL-PROTOTYPE SYS. LAB DEMONSTRATED	
COMMENT	SUCCESSFULLY F/T IN 680J PROGRAM		READY FOR FLIGHT TEST-ING FOLLOWING LIMITED QUAL TEST	

FIGURE A-2 ACTUATION SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	SERVO ACTUATOR SPERRY RAND CORP.	PITCH CONT SYS STUDY DOUGLAS AIRCRAFT CO AD-SKYRAIDER-A/C	SECONDARY ACTUATOR SPERRY PHOENIX CO.	ACTUATOR ELLIOTT BROTHERS OF ENGLAND
DESIGN CONCEPT	17-002	18-005	19-005	20-005
REDUNDANCY	FORCE SUMMING	SIGNAL SUMMING TORQUE SUMMING	ACTIVE FAIL PASSIVE	FORCE SUMMED
CONTROL	JET PIPE DIFFUSER SLOT	HYDR SERVO VALVES AC TORQUERS	JET PIPE	FLAPPER SERVOVALVE
POWER	HYDR.	HYDR	HYDR	HYDR
POWER SOURCE	TRIPLE ELECTRICAL & HYDRAULIC	N/A	N/A	N/A
ACTUATOR DESCRIPTION	TRIPLE A/C INPUT 2 STAGE, 4 WAY JET PIPE	TRIPLE REDUNDANT HYDR	DUAL	QUAD PARALLEL
FEEDBACK	INTERNAL HYDRAULIC	MECHANICAL	DUAL MECHANICAL	ELECTRICAL MECHANICAL
MONITORING	PARALLEL REDUNDANT POSITIVE SYNCHRON.	ELECTRICAL	NOT REQD PERFORMED FOR FAILURE REPORTING ONLY	HYDR/MECH
STATUS	DEVELOPMENT	PROGRAM REDIRECTED PROTOTYPE MODEL	MODEL LAB TESTED EXPERIMENTAL	TESTED DEVELOPED
COMMENT		NEGATIVE RESULTS		

ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	MC DONNEL DOUGLAS TRIPLE REDUNDANT FBW SYSTEM	4 CHAN-FORCE SUM SYNC SYSTEM BENDIX CORP	SECONDARY ACTUATOR SPACE SHUTTLE-HYDR RES & MFG CO.	ACTUATOR GENERAL ELECTRIC CO. F111 A/C
DESIGN CONCEPT	21-018	22-149	23-050	24-009
REDUNDANCY	SINGLE FAIL OPERATE	STANDBY	FO/FO/FS FORCE SUM	STANDBY
CONTROL	JET PIPE	E/M VALVE	FLAPPER NOZZLE	SERVOVALVE
POWER	HYDR	HYDR	HYDR	HYDRAULIC
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	DUAL TANDEM	SINGLE	TRIPLE TANDEM	DUAL TANDEM
FEEDBACK SENSOR	TRIPLE TANDEM LVDT	LVDT	QUAD-LVDT	MECHANICAL
MONITORING	POSITION	ΔP	HYDRAULIC + ELECTRONIC	HYDRO MECHANICAL
STATUS	LAB MODEL	CONCEPTUAL	ENGINEERING MODEL EXPERIMENTAL	OPERATIONAL ON F111
COMMENT	LAB TESTED			

ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	MAJ. VOTING SERVO ACTUATOR-MOOG INC F111 ACFT
DESIGN CONCEPT	25-092	26-092	27-092	28-123
REDUNDANCY	FORCE SUMMING	FORCE SUMMING	FORCE SUMMING	STANDBY
CONTROL	E/M VALVE	E/M VALVE	E/M VALVES	FLAPPER
POWER	HYDR	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	DUAL TANDEM	DUAL TANDEM	DUAL TANDEM	SINGLE
FEEDBACK SENSOR	LVDT	LVDT	MECHANICAL	MECHANICAL
MONITORING	POSITION	POSITION	POSITION	PRESSURE
STATUS	CONCEPTUAL	CONCEPTUAL	CONCEPTUAL	IN PRODUCTION
COMMENT				

FIGURE A-2 ACTUATION SUMMARY MATRIX

CLASS	ANALOG SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	ACTUATOR GENERAL ELECTRIC CO. A7A A/C	ACTUATOR GENERAL ELECTRIC CO.	POSN SUM SEC ACTUATOR - GE CO. 680J PROJ.	ELECTRO-RAM ACTUATOR LTV ARLINGTON, TEXAS
DESIGN	29-009	30-009	31-142	32-039
REDUNDANCY	FORCE SUMMING	FORCE SUMMING	TWO FAIL OPERATE	ACTIVE TWO-FAIL-OPERATE VELOCITY SUMMING
CONTROL	SERVOVALVE	SERVOVALVE	SOLENOID VALVES	BRUSHLESS A-C SERVO MOTORS
POWER	HYDRAULIC	HYDR.	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	DUAL PARALLEL	DUAL PARALLEL	N/A	DUAL TANDEM
FEEDBACK SENSOR	LVDT	LVDT	LVDT	TACHOMETERS
MONITORING	ELECTRONIC COMPARATOR	MAJORITY LOGIC COMPARATOR	POSITION	COMPARISON TACHOMETER SIGNALS
STATUS	OPERATIONAL ON A7A	FLIGHT TESTED	CONCEPTUAL	WILL FLY IN ADP 680J SFCS
COMMENT				

ACTUATION
FIGURE A-2 SUMMARY MATRIC

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	N/A
IDENTIFICATION	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	MOTORS LTV ELECTROSYSTEMS 680J PROJECT	AFCAS MODULAR ACTUATOR ROCKWELL INT., CAD
DESIGN CONCEPT	33-092	34-092	35-092	36-114
REDUNDANCY	DISPLACEMENT SUMMING	DISPLACEMENT SUMMING	VELOCITY/DISPLACEMENT SUMMING	N/A
CONTROL	SOLENOID OPERATED E/M VALVE	SOLENOID OPERATED E/M VALVES	E/M VALVE	DIRECT DRIVE CONTROL VALVE
POWER	HYDR	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	DUAL TANDEM	DUAL TANDEM	DUAL TANDEM	MODULAR
FEEDBACK SENSOR	LVDT	MECHANICAL	LVDT	LVDT
MONITORING	POSITION	POSITION	TACHOMETERS	N/A
STATUS	CONCEPTUAL	CONCEPTUAL	CONCEPTUAL	CONCEPTUAL
COMMENT				

ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/DIGITAL SERVO
IDENTIFICATION	FBW (SPOILER SYS) SPERRY PHOENIX CO. B-52H ACFT	ACTUATOR GENERAL ELECTRIC CO.	ACTUATOR SPERRY PHOENIX CO.	F-8 DIGITAL F-B-W
DESIGN CONCEPT	37-005	38-009	39-005	40-060
REDUNDANCY	SIGNAL SUMMING STANDBY	STANDBY	HYBRID ACTIVE/STANDBY	TWO FAIL OPERATE
CONTROL	SERVOVALVES	SERVOVALVE	SERVOVALVE	
POWER	HYDR	HYDR.	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	QUAD PARALLED	TRIPLE PARALLEL	TRIPLE PARALLEL	TRIPLE TANDEM
FEEDBACK SENSOR	MECHANICAL	LVDT	TRIPLE LVDT	
MONITORING	ELECTRICAL	ELECTRONIC COMPARATOR	POSITION MONITORED ELECTRICAL	HYDRAULIC PRESSURE
STATUS	CONCEPTUAL	DEVELOPMENT	CONCEPTUAL	F-8 FLIGHT TESTING
COMMENT				

ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	SECONDARY ACTUATOR SPACE SHUTTLE-HYDR RES & MFG CO.	SECONDARY ACTUATOR SPERRY PHOENIX CO	ACTUATOR SPERRY PHOENIX CO.	SERVO ACTUATOR SYS HYDRO RES & MFG CO. SPACE SHUTTLE
DESIGN CONCEPT	41-051	42-005	43-005	44-131
REDUNDANCY	FO/FO/FS ACTIVE/STBY	STANDBY TRIPLE REDUNDANCY	STANDBY	OPERATE/FAIL-OPERATE/ FAIL TRAIL
CONTROL	FLAPPER NOZZLE	ELECTRO HYD SOLENOIDS	SERVOVALVE	ELECTRO-HYDRAULIC SERVOVALVE
POWER	HYDR	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	TRIPLE TANDEM	DUAL	QUAD PARALLEL	HYDR COMPARATOR SELF MONITORING ACTIVE STANDBY
FEEDBACK	QUAD-LVDT	DUAL MECHANICAL	QUAD LVDT	LVDT
MONITORING	HYDRAULIC + ELECTRONIC	ELECTRICAL	POSITION MONITORED HYDR-ELEC	FAILURE DETECTION
STATUS	ENGINEERING MODEL EXPERIMENTAL	CONCEPTUAL	CONCEPTUAL	DEVELOPMENT
COMMENT	5% SWITCHING TRANSIENTS			

FIGURE A-2 ACTUATION SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	ACTUATORS LTV ELECTROSYSTEMS 680J PROJECT	REDUNDANT SERVO ACTUATOR-MOOG INC F111 ACFT	SERVO VALVES LTV ELECTROSYSTEMS 680J PROJECT
DESIGN CONCEPT	45-092	46-092	47-123	48-092
REDUNDANCY	ACTIVE STANDBY	ACTIVE STANDBY	DETECTION CORRECTION	ACTIVE STANDBY
CONTROL	E/M VALVE	E/M VALVE	SPOOL VALVE	E/H VALVE
POWER	HYDR	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	DUAL TANDEM	DUAL TANDEM	DUAL TANDEM	DUAL TANDEM
FEEDBACK SENSOR	LVDT	MECHANICAL	MECHANICAL FEEDBACK CAM	LVDT
MONITORING	POSITION	POSITION	ERROR SENSING	HYDR PRESSURE
STATUS	CONCEPTUAL	CONCEPTUAL	PRODUCTION F111	CONCEPTUAL
COMMENT				

ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO		
IDENTIFICATION	ACTUATOR SPERRY PHOENIX CO	ACTUATOR SPERRY PHOENIX CO		
DESIGN CONCEPT	49-005	50-005		
REDUNDANCY	STANDBY TRIPLE REDUNDANT	STANDBY TRIPLE REDUNDANT		
CONTROL	ELECTRO HYDR SOLENOIDS	ELECTRO HYD SOLENOIDS		
POWER	HYDR	HYDR		
POWER SOURCE	N/A	N/A		
ACTUATOR DESCRIPTION	TRIPLE REDUNDANT	TRIPLE REDUNDANT		
FEEDBACK SENSOR	ELECTRICAL	ELECTRICAL		
MONITORING	ELECTRIC SPOOL MONITORED	HYDR		
STATUS	CONCEPTUAL	CONCEPTUAL		
COMMENT	COMPLEX			

ACTUATION
FIGURE A-2 SUMMARY MATRIX

CLASS	SERVO/MECH. POWER ANALOG	VALVE DIRECT DRIVE ANALOG	100,000 IN-LB	N/A
IDENTIFICATION	INBD. ELEVATOR SERVO LOCKHEED-GA. CO. C5A TRANSPORT	PACIFIC CONTROLS MODEL 700 SERVO VALVE PACIFIC CONTROLS, INC	ROTARY ACTUATOR-DYNAVECTOR BENDIX CORP.	ROTARY ACTUATOR SPACE MISSIONS
DESIGN	51-109	52-110	53-023	54-024
REDUNDANCY	TRIPLEX SERVO DUAL/DUAL POWER ACT.	N/A	N/A	N/A
CONTROL	ELECTRO-HYDR. SERVO VALVE & MECH.	DIRECT DRIVE ELECTRICAL FORCE MOTOR	N/A	N/A
POWER	HYDRAULIC	ELECTRICAL	HYDR	HYDR
POWER SOURCE	ENGINE	N/A	N/A	N/A
ACTUATOR DESCRIPTION	DUAL/DUAL PWR. ACT. TRIPLEX SERVO	N/A	ROTARY	INTEGRATED MOTOR AND EPICYCLIC GEAR REDUCER
FEEDBACK SENSOR	L.V.D.T & MECH.	NOT SPECIFIED	N/A	N/A
MOTORING	ELECTRONIC	NOT SPECIFIED	N/A	N/A
STATUS	PRODUCTION	INDUSTRIAL AEROSPACE IN DEVELOPMENT	EVALUATION TESTED DEVELOPMENT	BREADBOARD MODEL DEVELOPMENT
COMMENT				

COMPONENTS
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	N/A	FLUID/SERVO	DIRECT DRIVE
IDENTIFICATION	PRI FLT CONT SYS F14, B58, F111 & 747 ACFT - BENDIX	MECH F/B ACTUATOR MOOG INC SPACE BOOSTER/IC BM	FLUIDIC SERVOACTUATOR & EM INPUT XDCR GE CO.	MISSILE FLIGHT CONT ACTUATOR CURTISS WRIGHT
DESIGN CONCEPT	55-149	56-123	57-151	58-152
REDUNDANCY	N/A	FAIL NEUTRAL	N/A	N/A
CONTROL	N/A	N/A	PUMP-BELLOWS FLAPPER-NOZZLE	SPRING CLUTCHES
POWER	HYDR	DIRECT DRIVE	FLUID	ELECTRIC
POWER SOURCE	N/A	N/A	N/A	BATTERY
ACTUATOR DESCRIPTION	MISC	N/A	N/A	N/A
FEEDBACK SENSOR	N/A	N/A	HYDR POSITION	LVDT
MONITORING	N/A	N/A	N/A	N/A
STATUS	IN PRODUCTION	IN PRODUCTION	LAB TESTED DEVELOPMENT	IN PRODUCTION
COMMENT				

COMPONENTS
FIGURE A-2 SUMMARY MATRIX

CLASS	TRANSDUCER	TRANSDUCER	POSITION TRANSDUCER	POSITION TRANSDUCER
IDENTIFICATION	INCREMENTAL ENCODER TELEDYNE GURLEY	CONVERTERS SINGER/KEARFOTT	TRANSDUCER G.L. COLLINS	D.C. TRANSFORMERS SCHAEVITZ
DESIGN CONCEPT	59-129	60-130	61-C	62-C
REDUNDANCY	WOULD REQUIRE TWO SEPARATE SYSTEMS, COMPUTER CONT.	WOULD REQUIRE TWO SEPARATE SYSTEMS, COMP. CONT.	FOUR LVDT'S ON COMMON PISTON	
CONTROL	REQUIRES ELECTRONIC TOTALIZING COUNTER & COMPUTER CONT.	COMP. -"READ COMMAND, OUTPUT IS 14 BIT BIN PARALLEL OR 13 BIT BCD		
POWER	5 TO 6 VDC DEPENDING ON UNIT USED	SYNCHRO - 400 HZ AC S/D CONV - +5 VDC, +15 VDC, -15 VDC	LESS THAN 1 VA PER UNIT	±15 VDC @ +20 MA
POWER SOURCE	DC POWER SUPPLY OPERATED FROM A/C 400 HZ OR 28 VDC	A/C 400 HZ SYSTEM POWER SUPPLIES FOR +5VDC, +15VDC, -15VDC	26V @ 400 HZ AC GEN OR CHOPPED DC	DC SUPPLY OR BATTERY
ACTUATOR DESCRIPTION	UNITS AVAILABLE FOR ROTARY & LINEAR APPLICATIONS	ROTARY INPUT TO SYNCHRO OR RESLOVER IS REQ'D	LINEAR ACTUATOR POSITION	
FEEDBACK	N/A	N/A	N/A	
MONITORING	COMPUTER	COMPUTER	ANALOG OUTPUT	
STATUS	PRESENTLY AVAILABLE	PRESENTLY AVAILABLE	OPERATIONAL MIL SPEC APPROVED	IN PRODUCTION
COMMENT				NOT PROVEN FOR AIRBORNE APPLICATIONS

COMPONENTS - SENSORS
FIGURE A-2 SUMMARY MATRIX

CLASS	TRANSDUCER	ENCODER SHAFT POSITION	TRANSDUCER	ENCODER, ROTARY, INCREMENTAL
IDENTIFICATION	DIFF. TRANSFORMER SCHAEVITZ	ENCODER CONRAD/ DATEX, CONTACTING SIZE 11	ENCODER CONRAC/DATEX SIZE 55	ENCODER CONRAC MODEL 23
DESIGN CONCEPT	63-C	64-C	OPTICAL SHAFT POSITION (ROTARY) ENCODER 65-C	66-C
REDUNDANCY				
CONTROL				
POWER	LESS THAN 1 WATT	CONTACTS RATED AT 3 MA PER BIT AT 15 VDC OR LESS		
POWER SOURCE	AC RVDT: 4 to 20 KHZ DC TVDT: ±15 VDC			±5 VDC @ 250 MA MAX
ACTUATOR DESCRIPTION				
FEEDBACK SENSOR				
MONITORING				
STATUS	IN PRODUCTION	MIL SPEC COMPLIANT	IN PRODUCTION	IN PRODUCTION
COMMENT	NOT PROVEN FOR AIR- BORNE APPLICATIONS	INTERNAL ELECTRONICS AVAILABLE	TEMP LIMITED PREVENTS MIL APPLICATION	NOT QUALIFIED TO MIL SPECS

COMPONENTS - SENSORS
FIGURE A-2 SUMMARY MATRIX

CLASS	TRANSDUCER	TRANSDUCER	TRANSDUCERS	TRANSDUCER
IDENTIFICATION	POTENTIOMETERS CONRAC 851 SERIES	ENCODERS KEARFOTT	SYNCHROS & RESOLVERS KEARFOTT	ENCODER-BRUSHLESS GENERAL
DESIGN CONCEPT	67-C	BRUSH ENCODER 68-C	69-C	HALL EFFECT TRANSDUCERS 70-C
REDUNDANCY	DUAL BRUSHES AVAILABLE			
CONTROL				
POWER	1.5W MAX			
POWER SOURCE	AC OR DC EXCITATION			
ACTUATOR DESCRIPTION				
FEEDBACK SENSOR				
MONITORING				
STATUS	IN PRODUCTION	OPERATIONAL	AVAILABLE	COMMERCIAL - DIGISENSOR WANDS - MICROSWITCHES
COMMENT	AC EXC. NOT REQ'D	MIL SPEC COMPLIANT		POSSIBLE REPLACEMENT FOR CONTACT & OPTICAL ENCODERS

COMPONENTS - SENSORS
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	FLY-BY-WIRE SYSTEM SPERRY PHOENIX CO CH46-VTOL AIRCRAFT	FLY-BY-WIRE SYS STUDY KAMAN H43B-HELICOPTER	FLY-BY-WIRE SYS. THE BOEING CO. X20 DYNASOAR	FBW (PITCH SYS) SPERRY PHOENIX CO. F111 ACFT
DESIGN CONCEPT	71-005	72-005	73-005	74-005
REDUNDANCY	ELEC SUMMING	STAND BY FAIL OPERATIONAL	TRIPLE REDUNDANT	STANDBY POSITION SUMMING (QUAD)
CONTROL	SERVO VALVES	SERVO VALVE	SERVO VALVE	SERVO VALVES
POWER	HYDR	HYDR	HYDR	HYDR
POWER SOURCE	N/A	N/A	N/A	N/A
ACTUATOR DESCRIPTION	REDUNDANT	DUAL HYDR.	DUAL	REDUNDANT
FEEDBACK SENSOR	ELECTRICAL		C*	QUAD ELECTRICAL
MONITORING	ELECTRICAL	ELECTRICAL SWITCH IN/SWITCH OUT	IN LINE ELECTRICAL	ELECTRICAL
STATUS	CONCEPTUAL	CONCEPTUAL	MOCK UP EXISTS NEVER FLOWN	CONCEPTUAL
COMMENT		LOW RELIABILITY		

SYSTEMS
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	N/A	N/A
IDENTIFICATION	FLY-BY-WIRE PFCS HONEYWELL, INC.	FBW-STAB, SPOILER & RUDDER SYSTEM B-1 ACFT	AUTOSTAB SYSTEM HAWKER SIDDELEY AVN HARRIER-V/STOL	VSTOL REACTION CONT SYS - HAWKER, ENGLAND HARRIER-V/STOL ACFT
DESIGN CONCEPT	75-043	76-113	77-160	78-160
REDUNDANCY	DUAL FAIL-OPERATION	FAIL-OPERATIONAL FAIL-SAFE	N/A	NONE
CONTROL			N/A	NOZZLE
POWER	HYDR	HYDR	HYDR	PNEUMATIC
POWER SOURCE	N/A	N/A	AIRCRAFT	ACFT ENGINE
ACTUATOR DESCRIPTION	DUAL TANDEM	DUAL TANDEM	N/A	N/A
FEEDBACK SENSOR		LVDT	NONE	N/A
MONITORING		SELF MONITORING	NONE	N/A
STATUS	CONCEPTUAL	DEVELOPMENT	IN PRODUCTION	IN PRODUCTION
COMMENT				

SYSTEMS
FIGURE A-2 SUMMARY MATRIX

CLASS	MECH-SERVO			
IDENTIFICATION	NOZZLE ACTR SYS HAWKER SIDDELEY AVN HARRIER-VSTOL ACFT			
DESIGN CONCEPT	79-158			
REDUNDANCY	N/A			
CONTROL	N/A			
POWER	AIR			
POWER SOURCE	ENGINE BLEED			
ACTUATOR DESCRIPTION	ROTARY-AIR			
FEEDBACK SENSOR	SERVO LOOP			
MONITORING	NONE			
STATUS	IN PRODUCTION			
COMMENT				

SYSTEMS
FIGURE A-2 SUMMARY MATRIX

CLASS	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO	ANALOG/SERVO
IDENTIFICATION	SIMPLEX PACKAGE LTV-ARLINGTON, TE F4 ACFT	INTEGRATED SERVO PUMP PACKAGE VICKERS DIV.	DUPLEX ACTUATOR PACKAGE GE CO. 680J PROJ.	TRIPLEX ACTUATOR PACKAGE GE CO. 680J PROJ.
DESIGN CONCEPT	80-026	81-027	82-105	83-105
REDUNDANCY	2 FAIL OPERATE	N/A	DUAL FAULT CORRECTION	TWO FAIL OPERATE
CONTROL	ELECTRO HYDRAULIC SERVOVALVE	ELECTRO HYDRAULIC SERVOVALVE	E/M VALVE	E/M VALVE
POWER	HYDR	HYDR	HYDR	HYDR
POWER SOURCE	NORMAL PLUS EMERGENCY	N/A	400 HZ 200V 3Ø MOTOR	400 HZ 200V 3Ø MOTOR
ACTUATOR DESCRIPTION	DUAL TANDEM	N/A	QUAD PARALLEL	DUAL TANDEM
FEEDBACK	LVDT	LVDT	LVDT	UNKNOWN
MONITORING	ELECTRICAL	PRESSURE	POSITION	POSITION
STATUS	GND & FLT TESTED EXPERIMENTAL	PERFORMANCE TESTED EXPERIMENTAL	CONCEPT ONLY	CONCEPT ONLY
COMMENT	INCLUDES AN INTEGRAL EMERGENCY HYDR SYSTEM			

IAPS
FIGURE A-2 SUMMARY MATRIX

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CONCEPT NUMBERING

KEY

CONCEPT 1-049 PULSE TRAIN DIGITAL SERVO SYSTEMS

TITLE

Item Find No.—
Corresponds to Literature Search Bibliography Find No.

A "C" in this location indicates the information was taken from a manufacturer's catalog

Concept Sequence No.

CONCEPT 1-107 DIGICON CONTROL VALVE

Process Systems Inc. (PSI) DigiCon valves are operated by direct digital control. Each DigiCon valve is actually made up of a cluster of smaller valves. Each of these smaller valve control elements is either on or off. Each is controlled by a separate binary on-off computer signal, and each is a different size.

If the cross-sectional flow area, or C_v, of the smallest valve control element is given a value of one, then the next smallest has value of two, then four, eight, sixteen, thirty-two, sixty-four, and one hundred twenty-eight.

For extra reliability, a DigiCon valve physically divides the two largest C_v values - 64 and 128 - into redundant sets of smaller elements each with a C_v value no larger than 32. Creating a typical eight-bit, twelve-element DigiCon valve.

Depending on which control element, elements, set or sets of elements is open or shut, the entire DigiCon valve is that much open or shut. And since the possible combinations of eight elements and element-sets - total 256, a DigiCon digital valve assures a positive control accuracy equal to 1/255 of the total C_v.

As a result, instead of a typical C_v controllability of plus or minus 2 or 3 percent like conventional analog control valves, DigiCon valves provide an inherent resolution of 0.39 percent.

More important, because flow rate is a function of C_v, squared, actual flow rangeability is even better. From a minimum flow rate value of 1. To a maximum of approximately 800 and a flow control accuracy of the critical higher flow rates of nearly 0.2 percent.

Further, since the multiple valve elements in the DigiCon design can each be opened or shut in 25 milliseconds, the entire DigiCon valve can slew from open to shut or to any precise position in between in 25 milliseconds.

With eight C_v control elements and element sets instead of one, the failure of one or more elements in either an open or shut position doesn't require operational shutdown.

In fact, since no single valve element in a 12-element DigiCon valve carries more than 12.5 percent of the valve's total flow capacity, a DigiCon valve operating in a typical flow control situation at under 87.5 percent of full capacity could conceivably experience failure of one of its largest valve elements and computer logic would override the failure to return the system to exact setpoint. At the same time, because a DigiCon valve's actuation system requires only unregulated low-pressure air instead of highly-regulated pneumatics or hydraulics, actuation is greatly simplified. And the possibility of valve failure is sharply reduced.

CONCEPT 1- 107 (continued)

Finally, because a DigiCon valve operates on direct digital command, the need for servos, accumulators, diaphragms, bellows, positioners, feedback pots, regulators and gauges is eliminated.

In the design of the DigiCon valve, see Figure A-3, fluid from the inlet pipe enters the inlet manifold (A) and arrives at the multiple control elements.

Depending on the computer command, some of these may be in the open position, while others are closed (B).

Passing through orifice control cages in the open elements, fluid then enters the coaxial discharge manifold (C) and exits through an outlet pipe. Because the DigiCon body design is coaxial, there are no restrictions on the cross-sectional area of the inlet manifold. Which means the manifold can be as large as required to minimize fluid velocity and erosion within the manifold. What's more, because flow through opposing orifices in each control element cage causes flow impingement and turbulence within the cage, and opposing sets of control elements cause further flow impingement and turbulence in the discharge manifold, DigiCon valves sharply reduce erosion, cavitation and vena contracta throughout. As an extra benefit, they virtually eliminate valve scream and rumble.

The opening and closing of each control element plug is achieved simply and reliably. In fact, the only tie between the computer and the valve is a single multiconductor electrical cable. The computer or digital controller sends parallel on-off signals via the cable directly to small 3-way solenoid valves (AA) housed in the valve manifold. When energized by an on signal, each solenoid valve then moves to admit unregulated air (BB) into the control element actuator assembly (CC). The element plug (DD), which has been held closed by a spring, is driven to a full open position by the pressurized air. And fluid flows through the open valve control element. To close the element again, the computer merely cancels its on signal to the solenoid, causing it to de-energize. De-energized, the solenoid valve reverts to its original position.

Shutting off the pressurized air source and allowing the pressurized air in the element's plug actuator to bleed away. The spring in the plug actuator drives the plug closed again and fluid flow halts.

Compared with conventional single-plug analog valves, the advantages of the DigiCon control element design are many. Because the DigiCon control plug has only two stable positions - open and closed - the DigiCon valve's response time is only limited by how fast the element's actuator can drive the plug from one position to the other. And since the control element plug in the inlet manifold is balanced against upstream and downstream pressure, and since the plug assembly has very little inertia, a DigiCon control element can shoot from closed to fully open and vice versa in as little as 25 milliseconds.

CONCEPT 1-107 (continued)

With no possibility of overshoot the DigiCon valve's unique balanced control plug design also means the valve can be opened against the full differential pressure for which it was designed. Again, because a series of DigiCon on-off plugs operated in a particular on-off configuration produce the exact same C_v every time, there's no dead band. Because the DigiCon plugs operate on the outside of the control element orifice cages instead of the inside, they're also self-cleaning. And because element plugs and seals are out of the high-velocity flow stream, plug and seal erosion is eliminated. In fact, valve elements tested to more than 3.5 million cycles are still bubble tight, with no visible sign of wear. Add to these features the high pressure drop capability provided by the valve's unique anti-cavitation design, and the rugged DigiCon valve can deliver significantly extended service without wear or major maintenance.

Moreover, when part changes or minor preventative maintenance become necessary, they're a lot simpler and easier. Modular valve element assemblies can be replaced in a matter of minutes. And since DigiCon valve components permit extensive interchangeability between valves of different size, trim and pressure rating, only a minimum inventory of spare parts is suggested.

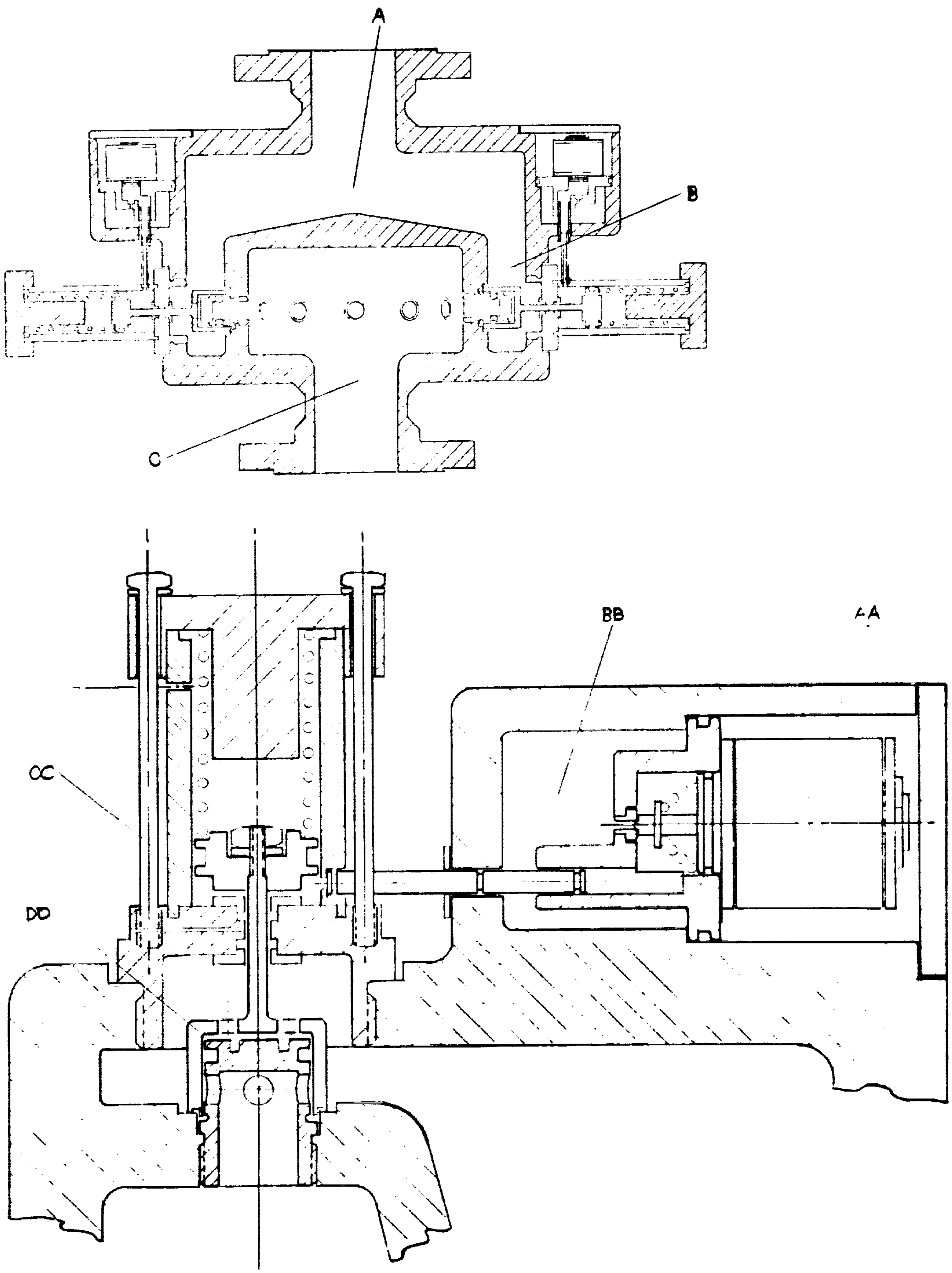

FIGURE A-3 DIGICON CONTROL VALVE

CONCEPT 2-161 ELECTRO-HYDRAULIC PULSE MOTOR

The FUJITSU electro-hydraulic pulse motor has been developed for use in the numerical control of machine tools, as well as after heavy industrial applications.

The electro-hydraulic pulse motor consists of an electric pulse motor and a feedback mechanism consisting of a four-way valve of rotary linear motion, a hydraulic motor, a screw on one end of the four-way valve spool, and a nut on one end by hydraulic motor shaft. It is in essense an electric pulse rotor and a hydraulic torque amplifier.

When the electric pulse motor rotates counterclockwise (as seen from the EHPM output shaft side), the four-way valve spool rotates clockwise and moves to the left (as illustrated in Figure A-4A). Consequently, the oil path is established on the + side of the four-way valve, so that the oil pressure is applied to the + side of the hydraulic motor, which results in rotation of the hydraulic motor clockwise. As the hydraulic motor rotates clockwise the nut portion of the motor shaft also rotates in the same direction, viz. the direction in which the valve spool initially rotated. Thus, the valve spool is brought back to its original neutral position, to close the oil path (as illustrated in Figure A-4B). In this manner, the output shaft of the hydraulic motor rotates in exact response to the rotation of the spool driven by the electric pulse motor.

From the electric pulse motor onward the EHPM is a closed-loop servo mechanism utilizing hydraulic pressure for torque amplification. In this portion, the oil path is simple and short, the inner oil volume is small, the inner leakage is extremely small, and the rotary member has a very small inertia. Thus, it serves as an ideal servo mechanism which has no overshoot and a very small dead band is shown in the characteristics diagram.

The work to be done by the electric pulse motor is limited to that of rotating the spool of the four-way valve. The load torque to be exerted by the electric pulse motor is limited to what is required for rotating the spool. Thus, the motor enjoys perfect freedom from the effect of the load applied to the hydraulic motor output shaft. This means that there is absolutely no possibility of the electric pulse motor functioning erroneously in consequence of an external disturbance even when there occurs a variation in the load on the part of the machine tool. Figure A-4C shows a cross-section of the pulse motor.

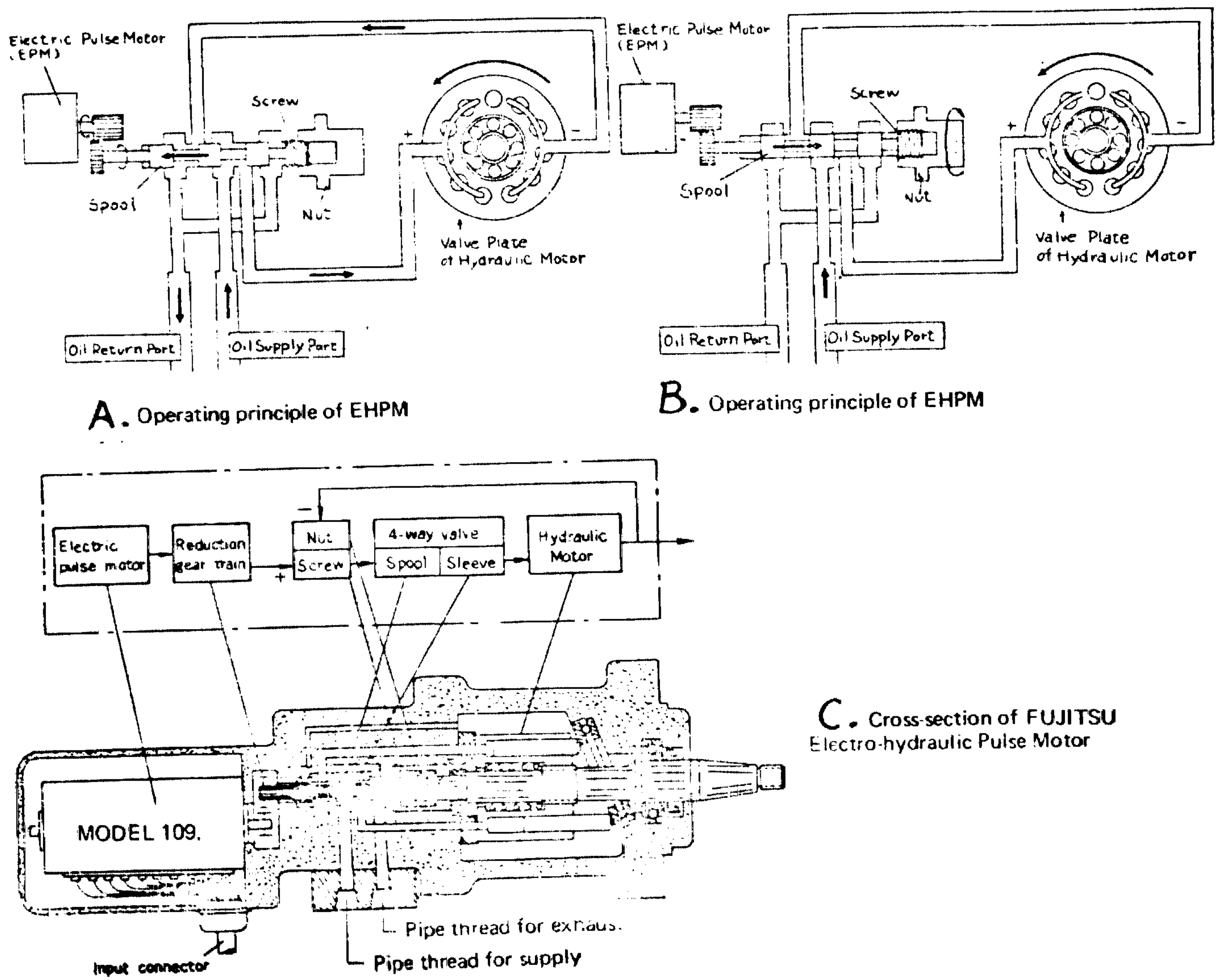

FIGURE A-4 ELECTRO HYDRAULIC PULSE MOTOR

CONCEPT 3-166 PULSE MOTOR WITH HYDRAULIC HELIX COUPLING TO PILOT VALVE CONTROLLED POWER SPOOL

In this approach a pilot hudraulic spool is placed between the pulse motor and power spool. See Figure A-5.

In essence the device consists of a pulse motor, a pilot spool with a coaxial power spool, and a valve body. The pulse motor is directly coupled to the pilot spool through a zero backlash coupling (such as bellows, etc.). The pilot spool is axially restrained between a pair of needle thrust bearings (both lubricated by the surrounding hydraulic fluid). Thus, the pilot spool accepts only rotary input from the pulse motor. A helical land on the pilot spool nulls between a pressure port and a return port in the coaxial power spool. A control pressure (C) is created in the helical groove and ported to an annular area (Ac) on one side of the power spool. System pressure ported to an annular area (Ap) on the opposite end of the power spool. System pressure acting on one side of the power spool is balanced by control pressure on the opposite. The power spool must be free to translate axially but must be restrained from rotation. This can be accomplished by providing an adjustable conical pin in the valve body which would engage in a close fitting slot in the power spool outside diameter. Adjusting the position of the pin would provide the necessary clearance without contributing significant backlash. The only load to burden the pulse motor is friction due to rotating the pilot spool in the power spool bore, at the needle thrust bearings, and at the low return pressure shaft seal; all very low. As previously described, the only external inertial load on the pulse motor is the pilot spool (and its coupling to the motor).

The power spool would have four null surfaces as in a conventional servovalve. The hydraulic follower (pilot helix) would have two null surfaces. There would be no steady state pilot stage flow other than leakage past the helix. Indexing the zero flow spool position to the zero signal pulse motor position would be accomplished at the shaft coupling. The coaxial design permits a compact package made up of a few precise parts as in a servovalve. No orifices or nozzles are required. The mecahnical design due to its few parts, lack of orifices, and few seals as well as its low motor burden, should provide a high mechanical reliability. The mechanical parts are few, though precise, and if properly tooled should provide a relatively economical design. This is not the whole picture, however. While this particular scheme provides the minimum mechanical hardware (with the possible exception of servovalves), all pulse motor drive schemes require the most complex electrical black boxes.

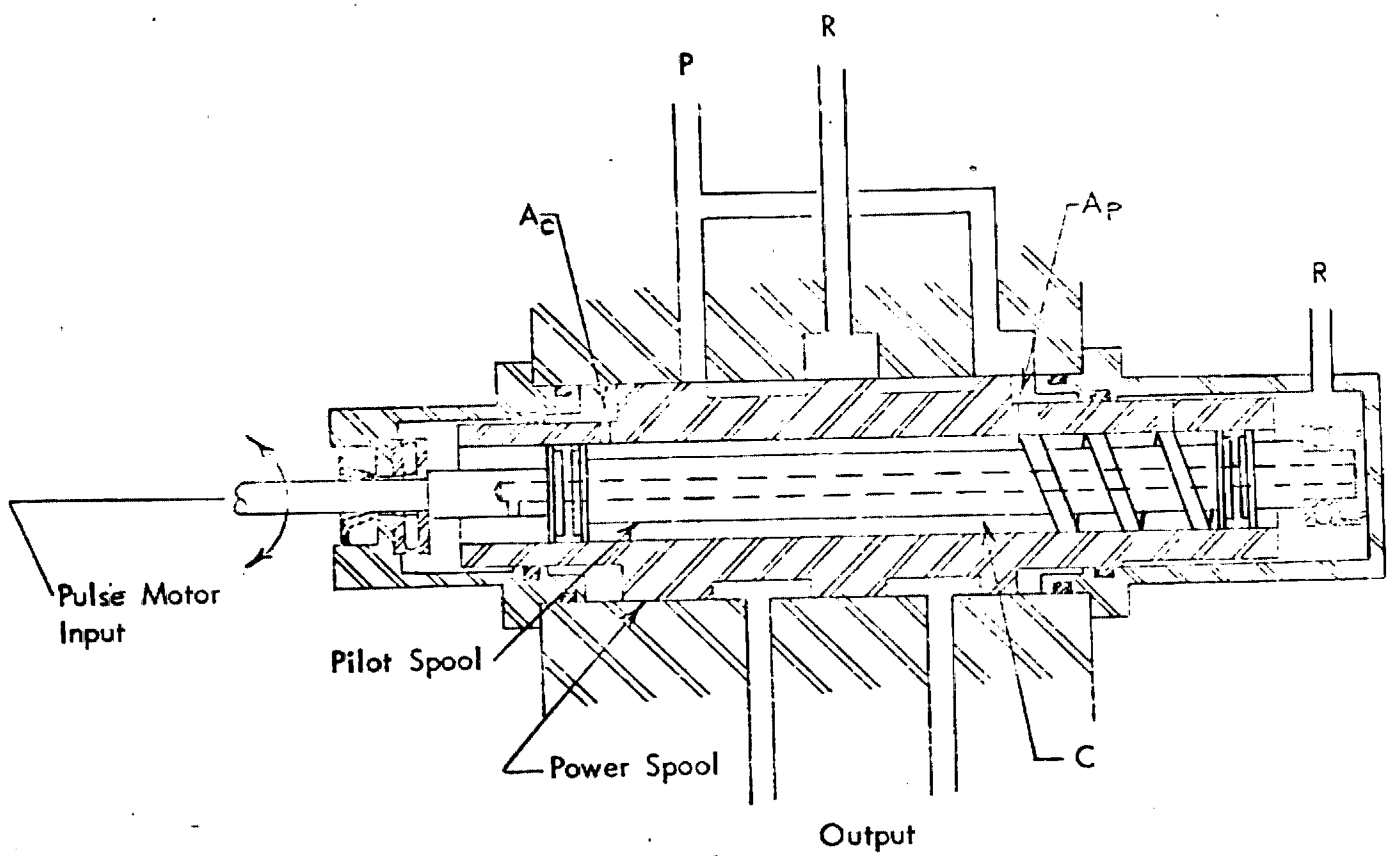

FIGURE A-5 PULSE MOTOR WITH HYDRAULIC HELIX COUPLING TO PILOT VALVE CONTROLLED POWER SPOOL

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CONCEPT 4-049 ELECTRO HYDRAULIC COMPONENT FOR A PARALLEL BINARY INPUT

A basic mechanization approach to an electrohydraulic component for a parallel binary input is shown in Figure A-6.

A servo valve first stage serves as a simple digital-to-analog converter. There are eight separate valve coils of differing number of turns and each coil has double the turns of the coil used for the adjacent bit. The total number of turns of all the coils is equal to the number of turns of a single coil which would be used in an analog version of the servo valve. Thus, the torque motor size is unchanged.

The continuous parallel binary input operates static switching circuits, applying voltage to the individual coils whenever an "on" stage exists in their respective channel. The most significant digit (MSD) coil has 128 N turns, while the least significant digit (LSD) coil has N turns.

A trimming resistor is placed in series with each coil and adjusted so that when the coil is energized, it exerts a torque of proper magnitude in relation to the other coils. The sum of the torques applied to the armature is cancelled out by the torque from the position feedback spring thus causing the actuator to move to the position represented by the binary input.

The use of trimming resistors provides the same current through each coil so the torque is proportional to the number of turns in the coil. Coil winding errors may be compensated by adjusting the trim resistors so that the final torque applied to the armature has the correct magniture. The total electrical power consumption when all coils are in the "on" state will be eight times that of the MSD channel, which itself requires one-half the power for the coil of an analog type valve.

The single coil uses the concept in its simplest form by controlling current flow through only one coil. This can be accomplished by use of a Kirchoff adder. A schematic of this variation is shown in Figure A-7. Each of the eight resistors shown binary weigh the current flowing through them. This current is summed at the indicated node. The binary weighted current next flows through a single coil whose resistance is negligible with respect to the MSD resistor. Thus, as in the multicoil version, an actuator displacement is produced proportional to the binary word defined by the on or off state of the resistor circuits. Since the resistance in the torque motor coil circuit will be dependent on the magnitude of the binary word inserted, the torque motor coil circuit time constant will again be amplitude sensitive.

Advantages and Disadvantages

Some advantages of the parallel binary approach are as follows:

- It will accept signals directly from a digital computer with no additional equipment required other than a hold circuit.
- The mechanical actuator feedback will cause the actuator to return to null in the absence of an input signal.

CONCEPT 4 -049 - (continued)

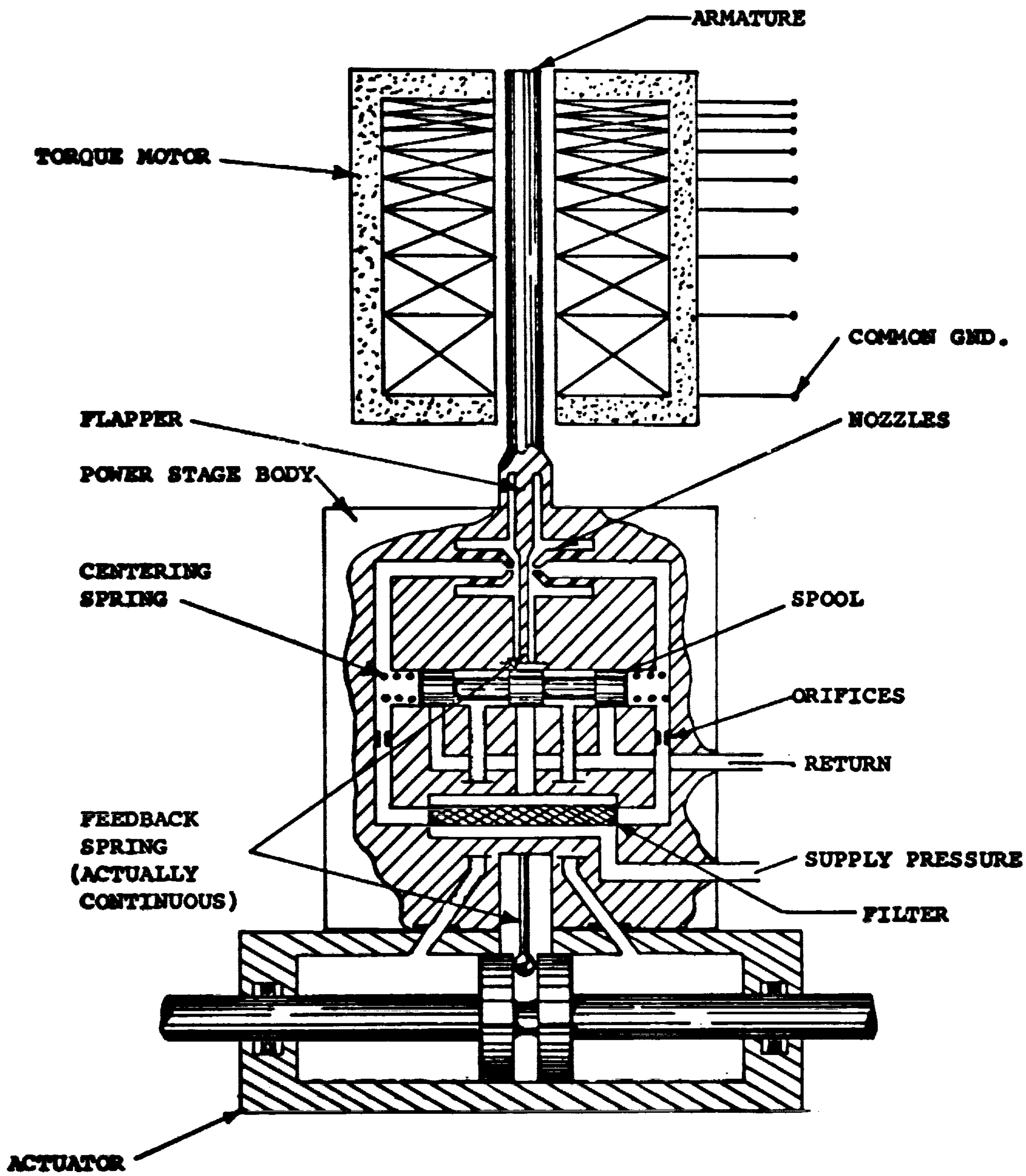
- The hardware is almost identical with conventional servo actuator design. Hence, little development difficulty will be encountered and initial reliability will be high.

Some disadvantages of this approach are as follows:

- Inherent in the multiple coil version of the parallel binary approach is the requirement for a wire between the computer and the valve for each channel. This could present a wire routing problem in the vehicle.

- Since the hardware is essentially the same as conventional servo valve components, this approach retains most of the disadvantages of conventional servo valves. Two of these are quiescent leakage and contamination sensitivity.

- Actuator feedback is still definitely required.



FIGURF A-6 PARALLEL BINARY BREADBOARD SCHEMATIC

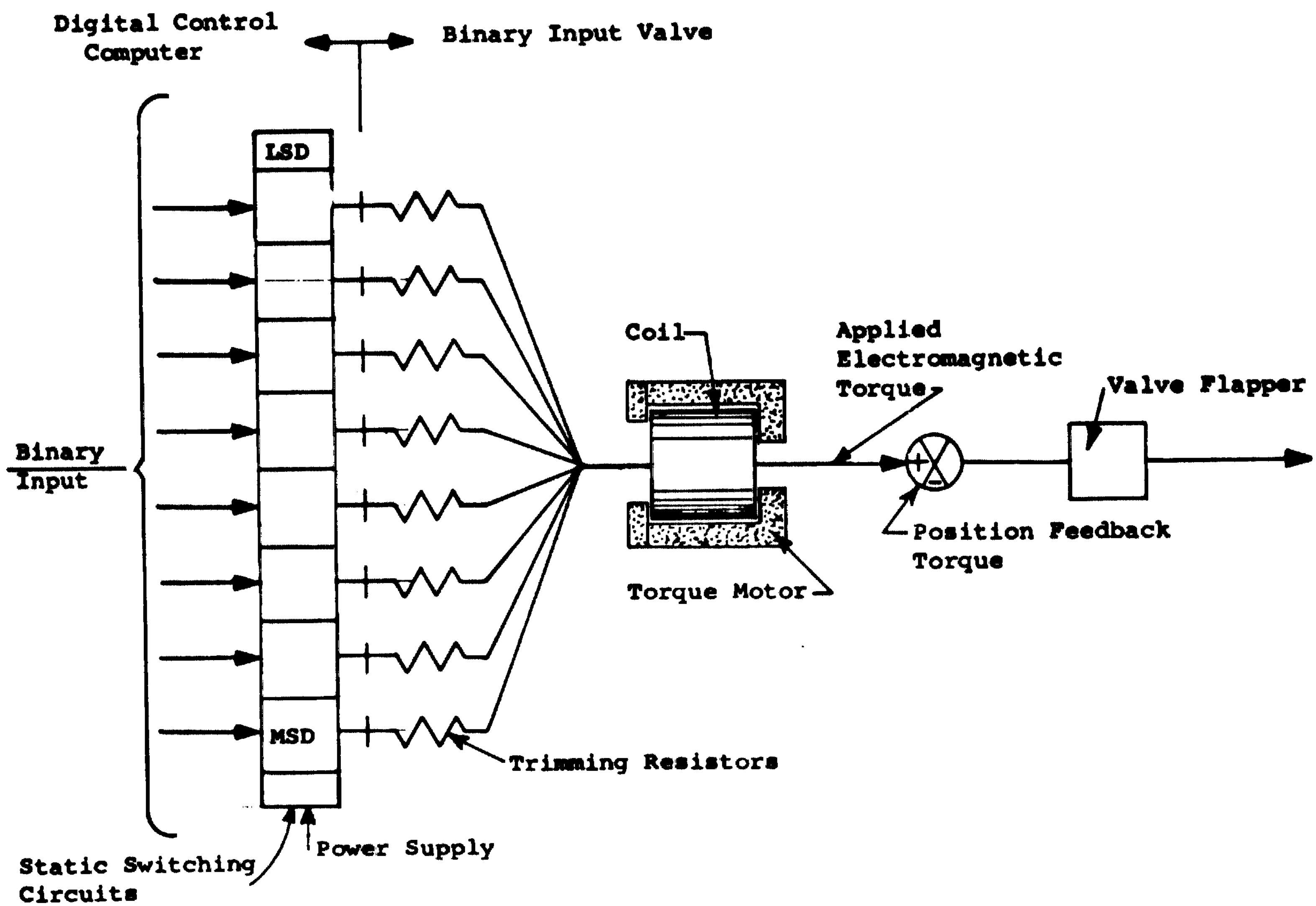


FIGURE A-7 BINARY INPUT SCHEMATIC (SINGLE COIL)

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CONCEPT 5-082 - DIGITAL SERVOVALVE (DSV)

The Digital Servovalve (DSV) is a development of the Hydraulic Research and Manufacturing Company, Valencia, California, for digital control of a missile launcher postion control system. The DSV provides flow output proportional to ± 7 bit parallel binary two's complement input signals.

The significance of the DSV is that it replaces the combination of an electronic D/A converter and analog servovalve with the intent of an improved output fidelity to command inputs.

The D/A mechanization, as shown in Figure A-8 consists of a standard Hydraulic Research second stage mechanical feedback servovalve driven by a digital torquer operating in the area of torque output saturation. The torquer consists of eight solenoidal bits, seven of which are positive torque output devices and the eighth most significant bit is negative. The mechanical summation of torques is accomplished by applying the array of torques to a common rotor. The rotor also receives feedback torque from second stage position which drives the summation of torques to zero.

The DSV has demonstrated a potential for static performance equivalent to state-of-the-art analog electrohydraulic servovalves. A 20-word per second input rate capability and less than 1-percent resolution was demonstrated. These results are significant when the attained power control is considered. The experimental model performed the digital-to-analog modulated control of 0 to 9 hydraulic horsepower.

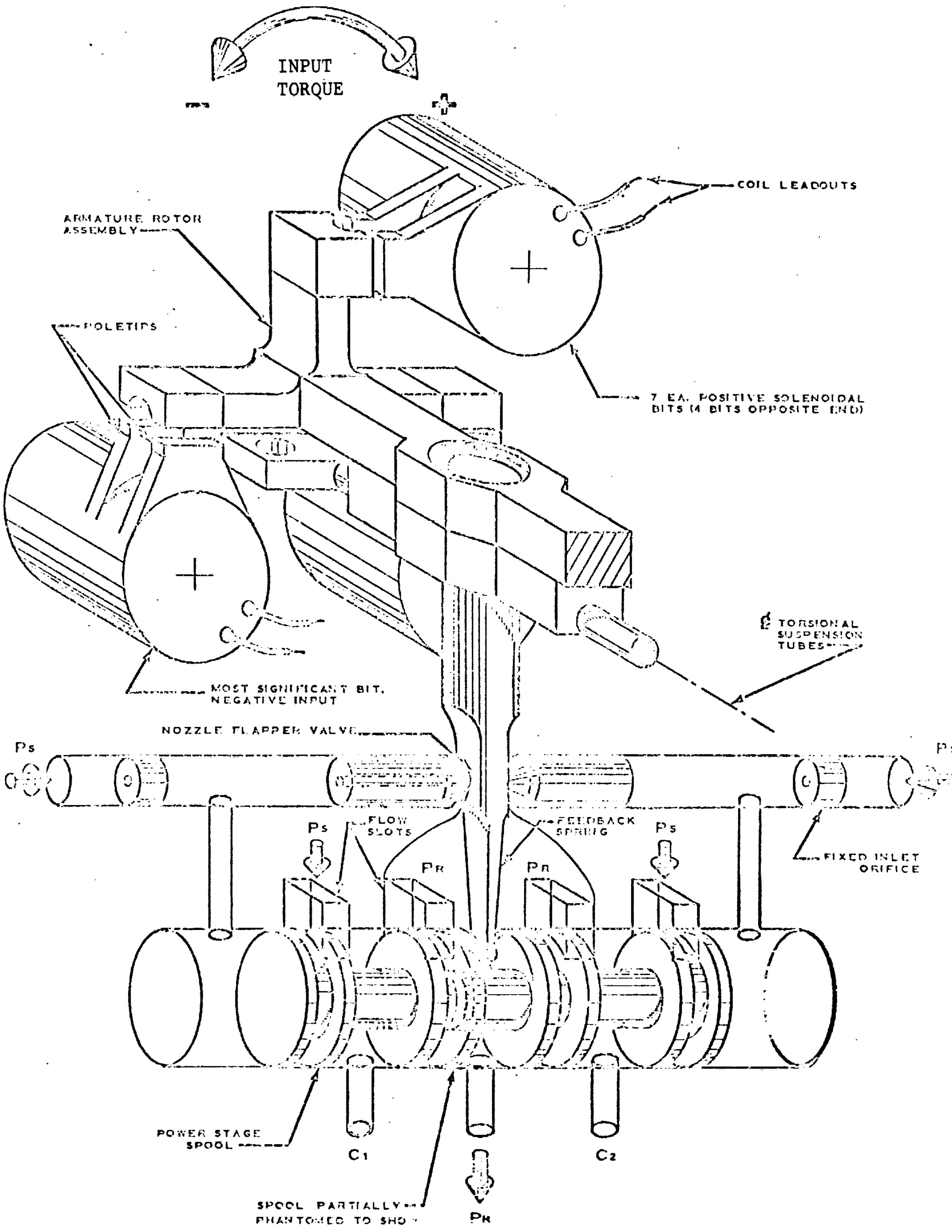

FIGURE A-8 DIGITAL SERVOVALVE OPERATIONAL SCHEMATIC

CONCEPT 6-061 HYBRID DIGITAL SERVO

The basic objective of the hybrid digital servo is to realize a practical digital power servo without the use of a complex digital servo actuator. A fundamental requirement of such a servo is that it operates on signals from the digital computer with a minimum of interfacing electronic equipment either interior or exterior to the computer itself.

The approach that has been followed in the development of this hybrid digital servo is to use a conventional analog electrohydraulic servo valve as a digital-to-analog converter; this technique quantizes valve flow rate and actuator velocity. By use of an encoder on the hydraulic actuator, a digital position feedback signal is obtained for closure of the servo loop.

In this mechanization all electrical signals are digital, but the hydraulic devices are conventional analog components. This is accomplished by modifying the electrohydraulic servo valve so that it will accept digital signals. In order to make the adaptation, the valve torque motor coil must be wound in several segments--four, for example, as shown in Figure A-9. Each coil segment is driven by a separate current source that is switched on or off at the command of the servo loop position error signal.

The coil segments are driven by a parallel binary digital signal that is "held" on the valve to command actuator velocity. The number of turns in each valve coil is proportional to the "weight" of the lowest order binary digit to which it is connected. For example, if the resolution coil (2^0 coil) contains 150 turns, the next three higher order coils will contain 300 turns, 600 turns, and 1200 turns. The power switches energize each coil with equal current, and the resulting ampere-turns of the coils are proportional to the weights of the binary digits. Since valve flow rate is proportional to the sum of the ampere-turns in the torque motor, the valve flow rate is quantized by the binary-weighted ampere-turns.

The flow-rate quantum level is determined by the number of coils in the valve. Through the use of four coils, and with positive and negative signals, it is possible to command 30 levels of valve flow rate other than zero--15 positively- and 15 negatively-phased flows (actuator extending and actuator retracting). Any position error signal equal to or larger than four binary bits will cause valve saturation.

Given a sampled serial command signal and a servo actuator that quantizes actuator rate when commanded by a parallel digital signal, the remaining task is to close the servo loop. Two basic digital techniques for electrical loop closure are considered feasible. These employ either a whole-word position encoder or an incremental position encoder.

Parallel Feedback

The schematic of this system is given in Figure A-9 which shows a serial command signal and a parallel feedback signal with the difference between the two applied in parallel to the four valve coils. The loop input signal is updated at the iteration period programmed into the computer, perhaps every 0.030 second, and must be held in an input position register. The feedback transducer signal is sampled at more frequent intervals to achieve the desired actuation dynamic characteristics.

Incremental Feedback

The block diagram for one of the possible configurations for incremental feedback servos is given in Figure A-10. In this example the actuator position is determined from the state of the storage register that is used to accumulate feedback increments.

It is also feasible that the computer command "change of actuator position". In this case, the command is shifted into an up-down counter to which the valve coil drivers are connected; feedback pulses drive the counter down to zero position error. The local loop is closed around an incremental command for change of actuator position. Other combinations of these techniques are also possible.

Redundancy

The operating principle of the hybrid digital servo provides the system with an inherent degree of redundancy. Provided the position feedback path remains intact, positional calibration is retained despite failures in torque motor coils, power switches, or logic units. Given a command, the servo provides the correct position for that command, regardless of these failures. Its dynamic characteristics, however, do change.

The velocity quantization characteristics are 15 regular velocity steps obtained with 15 units of position error. If the 2° coil is inoperative (open coil, failed power switches, etc.), the step size is double that of the "no failure".

As another example, if the 2^2 coil is inoperative, velocity characteristics are intact through the first three steps, but go to zero at four error units. (The 2^2 coil provides four velocity units). From four through seven error units the velocity is reduced by four units from its normal value. Between eight and eleven error units, velocity returns to its normal schedule which is repeated between twelve and fifteen error units. Note that the maximum velocity has been reduced to eleven units in this failure mode.

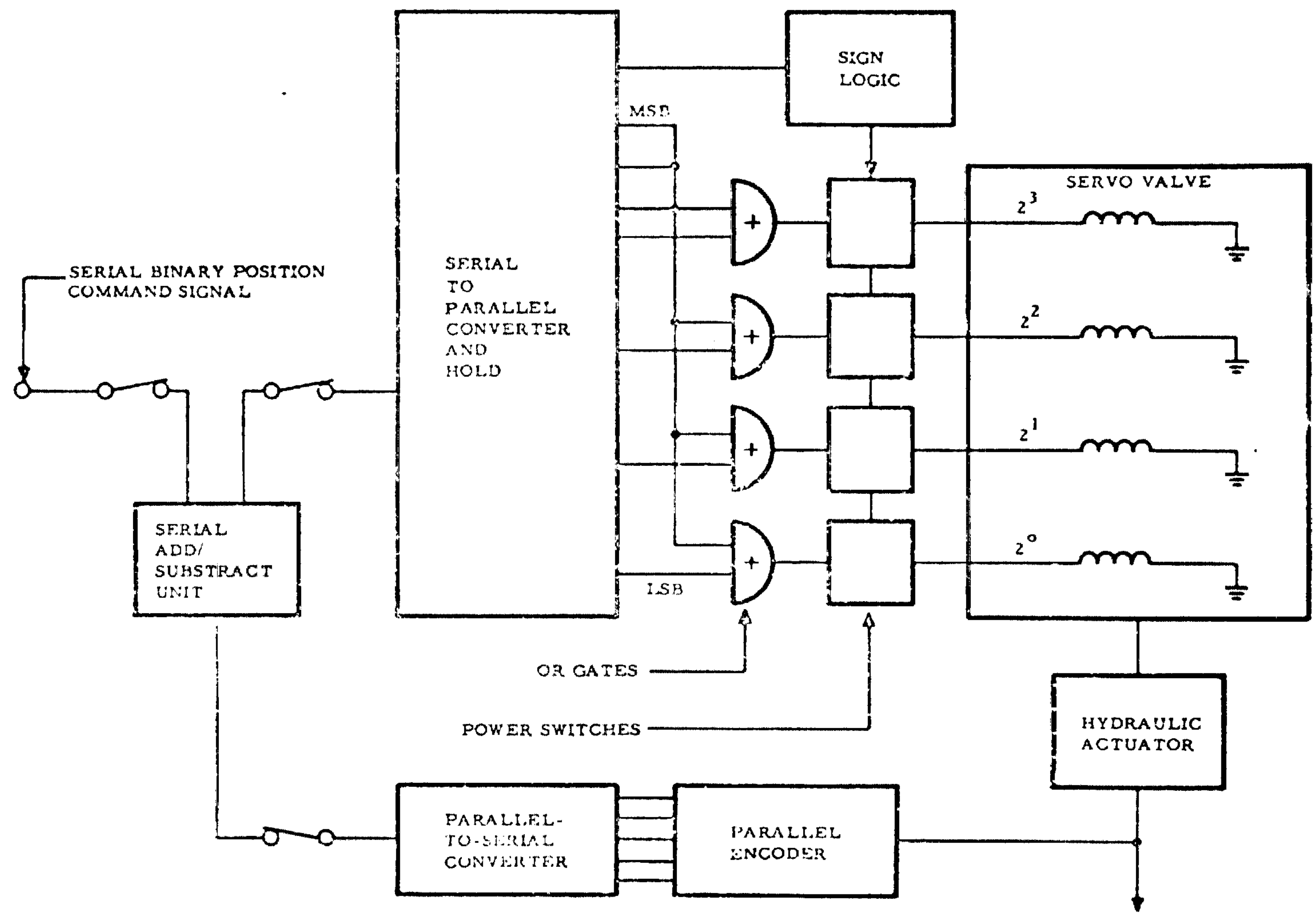

FIGURE A-9 DIGITAL SERVO WITH PARALLEL FEEDBACK

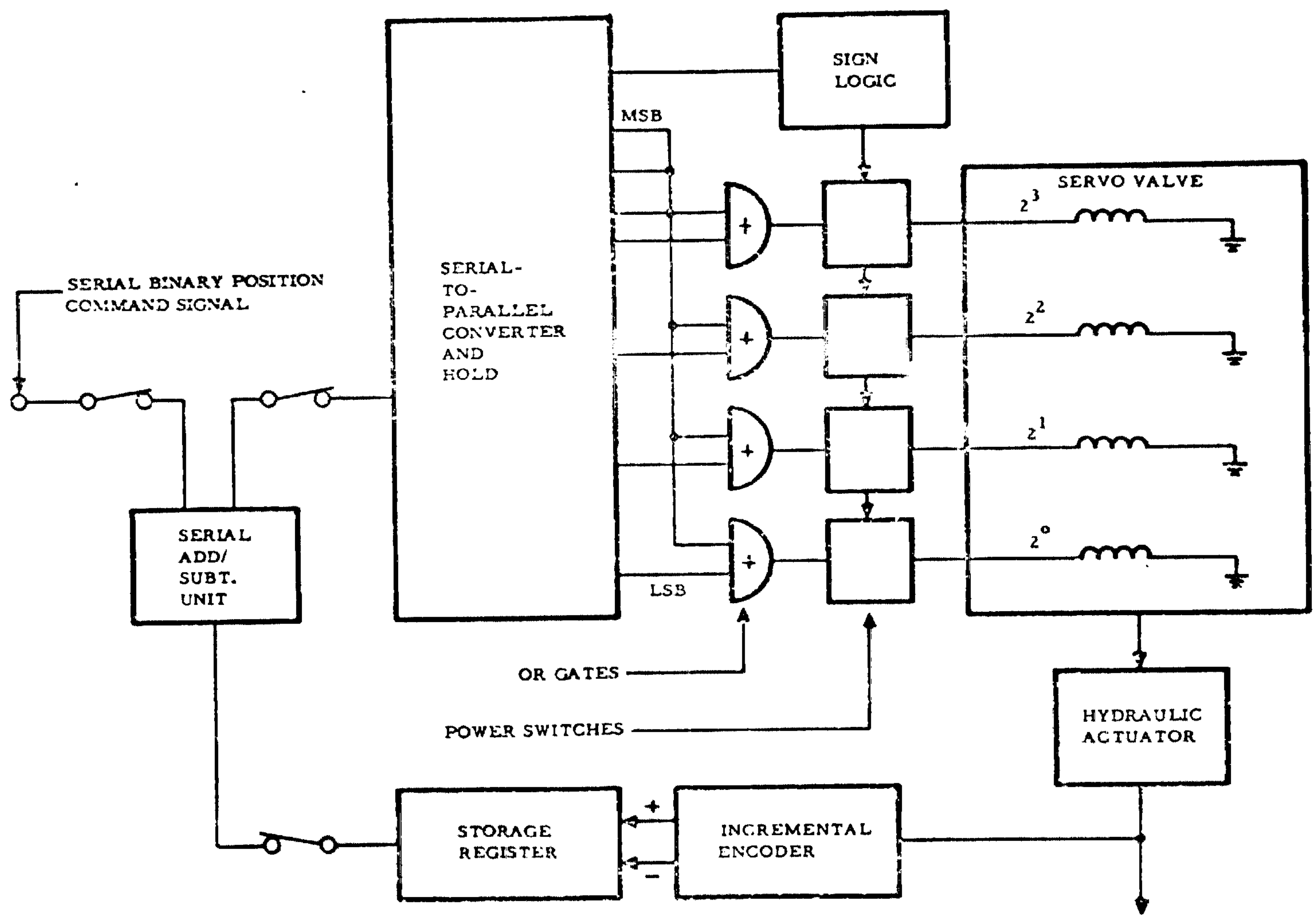

FIGURE A-10 DIGITAL SERVO WITH INCREMENTAL FEEDBACK

CONCEPT 7 - 168 PIEZOELECTRIC FLAPPER SERVOVALVE

Figure A-11 shows the actual construction of the flapper.

In order to obtain a large displacement, the cantilever consisting of "Bimorph" piezoelectric slabs is employed. The displacement of the free end of the cantilever is used as an input signal of the newly devised nozzle-flapper valve. In order to obtain the high natural frequency of the flapper, the bimorph piezoelectric slabs are bonded onto the both sides of a steel slab as shown in Figure A-11. The steel slab strengthen the flapper against a mechanical shock. The oil jets from nozzles do not directly strike the piezoelectric slabs as shown in Figure A-12.

When an input e(t) is applied to the modulator, the signal is converted into the form of a pulse train which actuates the flapper. The back pressure difference between the nozzle 1 and the nozzle 2 arises from the flapper displacement according to the pulse train, and this pressure difference results in the spool displacement. Briefly speaking, the average spool poisition is proportional to the input e(t), and the spool oscillates with a small amplitude and the frequency equal to that of the carrier wave.

The main parts of the servovalve are the piezoelectric flapper, the two nozzles, and the spool valve loaded by two feather springs. This modulator consists of a saw-tooth wave oscillator and a comparator.

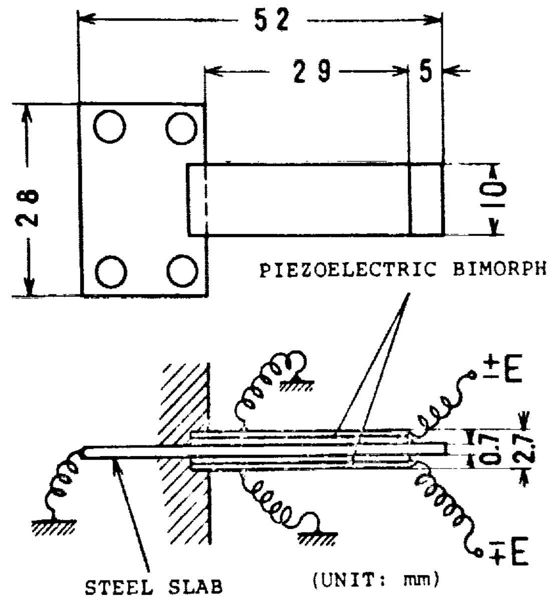

FIGURE A-11. CONSTRUCTION OF THE PIEZO-ELECTRIC FLAPPER

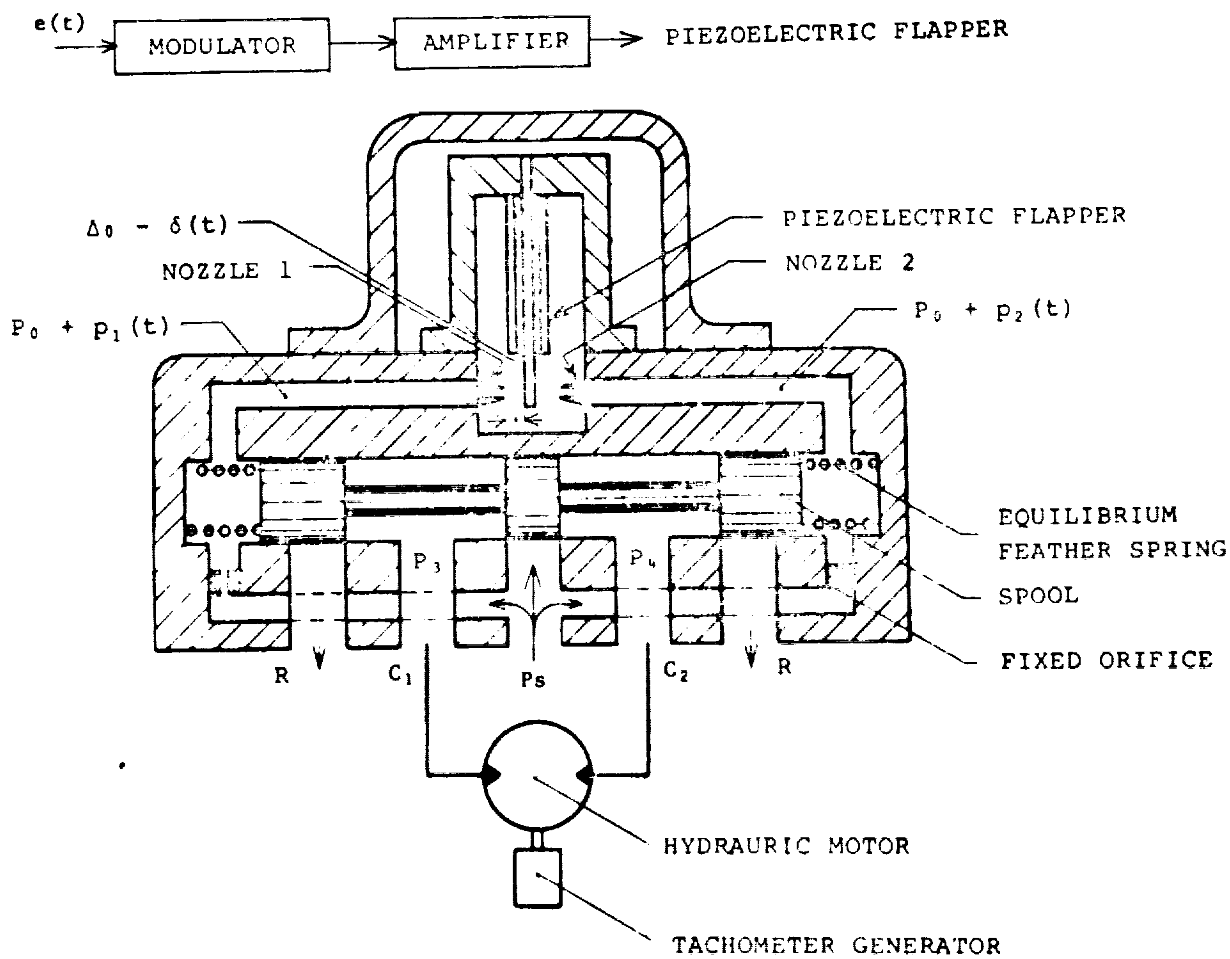

FIGURE A-12. ILLUSTRATION FIGURE OF THE NEWLY DEVISED SERVOVALVE

CONCEPT 8 -173 DIGITAL ACTUATOR

In this concept, Figure A-13, coils 1 and 2 receive pulse wave modulated (PWM) positional command signals and control the flapper position. In the null position, the flapper is centered and equal pressure appears at the nozzles and at both ends of the spool valve. When the flapper becomes influenced by a coil, the respective nozzle flow is restricted which increases the pressure on the end of the spool causing it to translate and direct system pressure to the actuator. A mechanical follow-up and summing linkage is attached directly to the actuator and drives a transducer which provides the positional follow-up signal back to the electronics thus closing the loop.

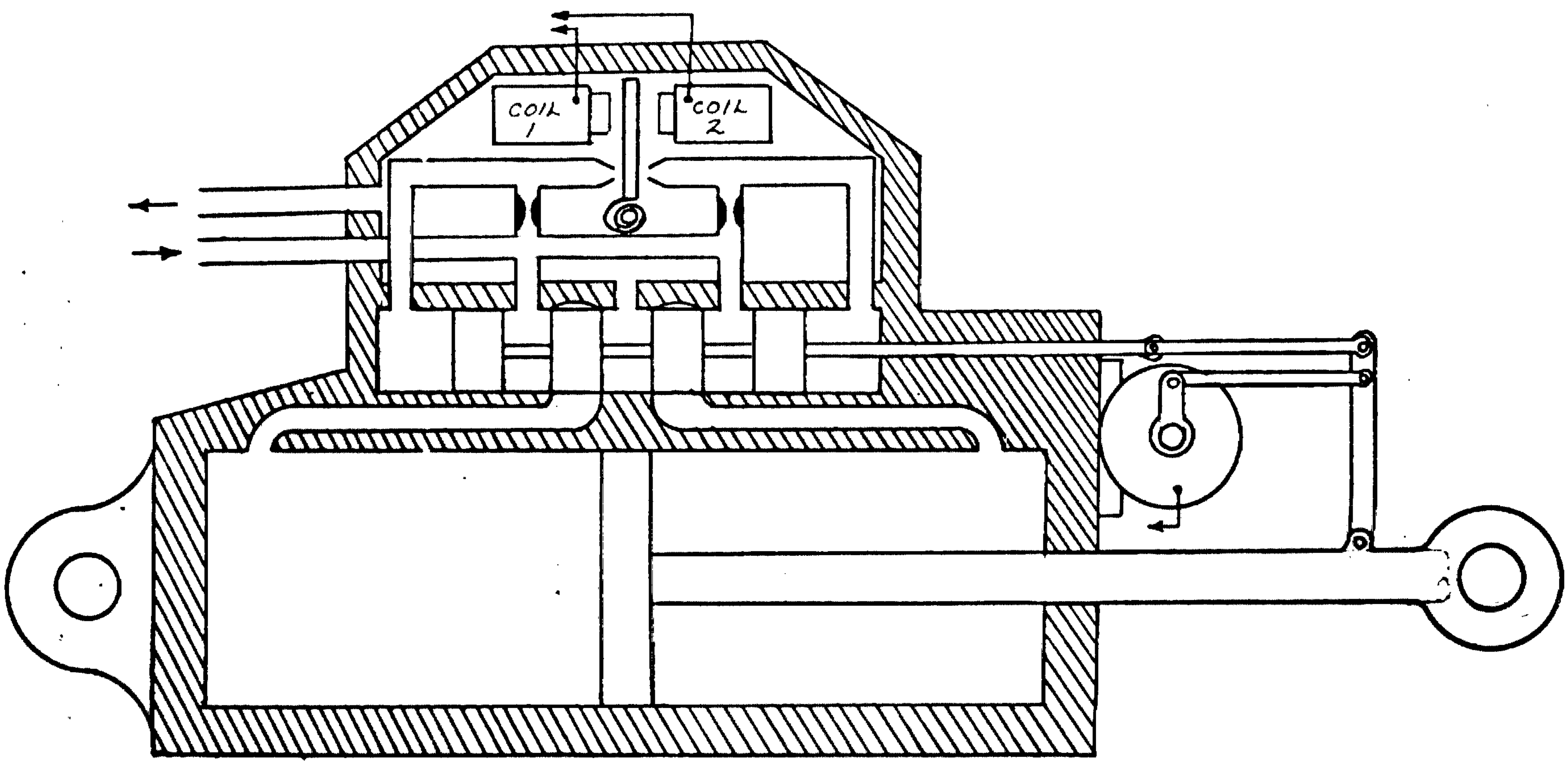

FIGURE A-13 DIGITAL ACTUATOR

CONCEPT 9-049 PULSE TRAIN DIGITAL SERVO SYSTEMS

The two-way shuttle system shown in Figure A-14, was designed to operate on incremental digital signals. Polarity is determined by current direction or coil selection resulting in an actuator position which is at any instant, the algebraic sum of the signals received to that time.

The torque motor and pilot section shuttle the polarity and sequencing spools. The polarity spool vents the oil to one side or the other of the actuator depending upon the polarity of the initial signal into the torque motor. The sequencing spool cycles the digitizer piston which meters a calibrated volume of oil to the actuator.

Advantages:

- Signal integrity is not important. As long as a pulse of minimum magnitude and duration is received the unit will function.
- The "bang-bang" type of operation eliminates the quiescent leakage and contamination sensitivity of the conventional servo valve.
- Conventional type components and design are used throughout.

Disadvantages:

- Response limitations of torque motors would limit the actuator slew rate.
- The system does not have the capability of driving the actuator to null in the event of electrical power failure or shut down.

In Figures A-15 and A-16 , two other mechanization approaches for pulse train type digital servo systems are shown. In all three systems each of the two spools has a separate function to perform. The difference is that the polarity spool varies its function according to which direction the spools are shuttled while the sequencing spool must perform exactly the same function regardless of the direction. The objective is to perform these functions with a minimum of component complexity and porting. In the system shown in Figure 2.2-3 the attempt was made to limit the sequencing spool travel to one direction no matter which polarity signal was put into the torque motor. This system requires twice the volume of oil to fill the cavity between the "split" spools as is required on the right end of the polarity spool for the "both spools left" position. This produces complicating unbalance conditions. In addition, the orificing system would be complex and would present contamination sensitivity problems.

CONCEPT 9-049 - (continued)

The system shown in Figure A-16 accomplishes the basic requirements by having each spool handle the polarity and digitizer piston cycling functions for its own digitizer piston.

This system has two major faults. First is the reliability disadvantage of adding the second digitizer piston. Second is the difficulty of matching the two digitizer calibrated volumes. Any miscalibration would produce actuator position errors.

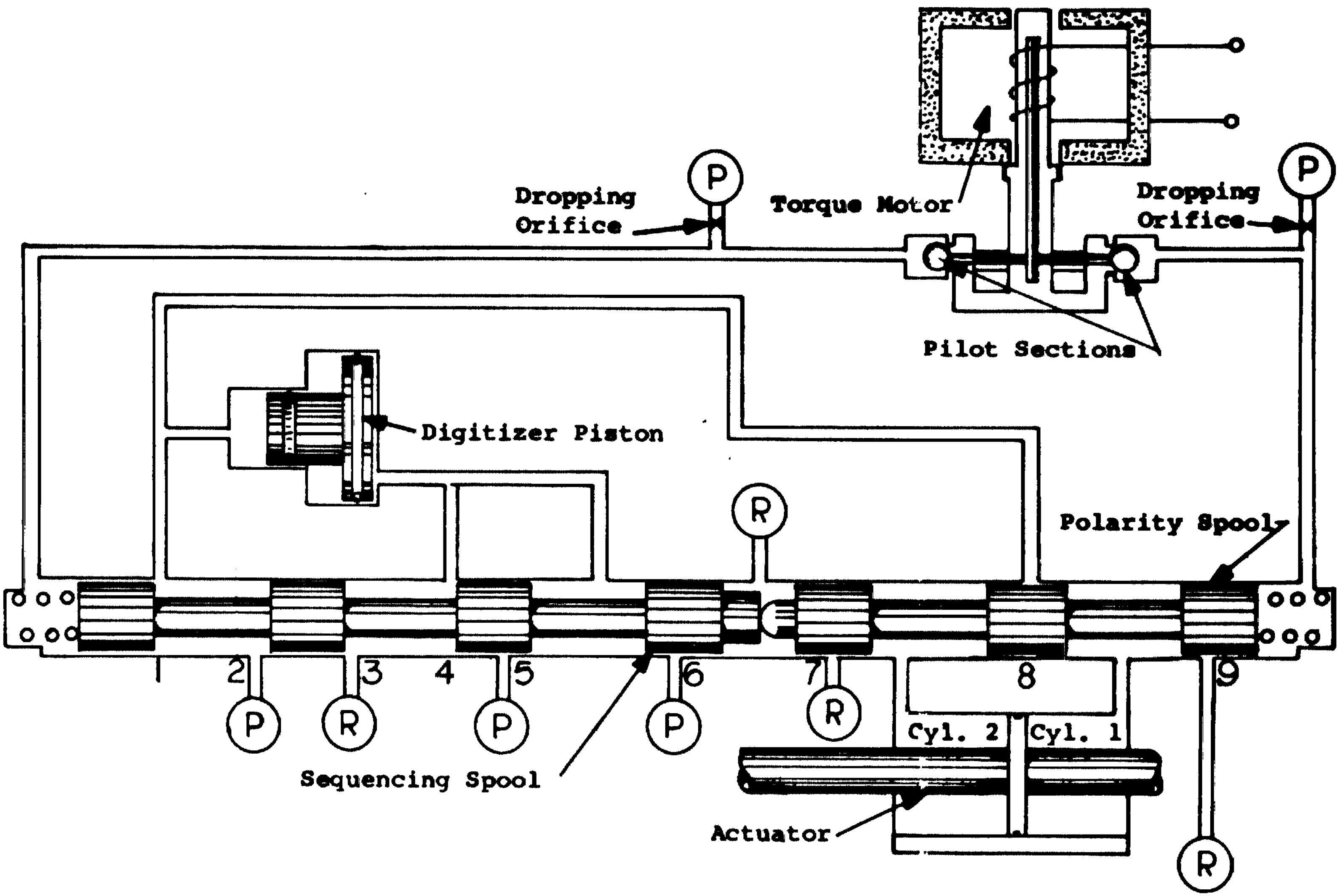

FIGURE A-14 TWO-WAY SHUTTLE SYSTEM

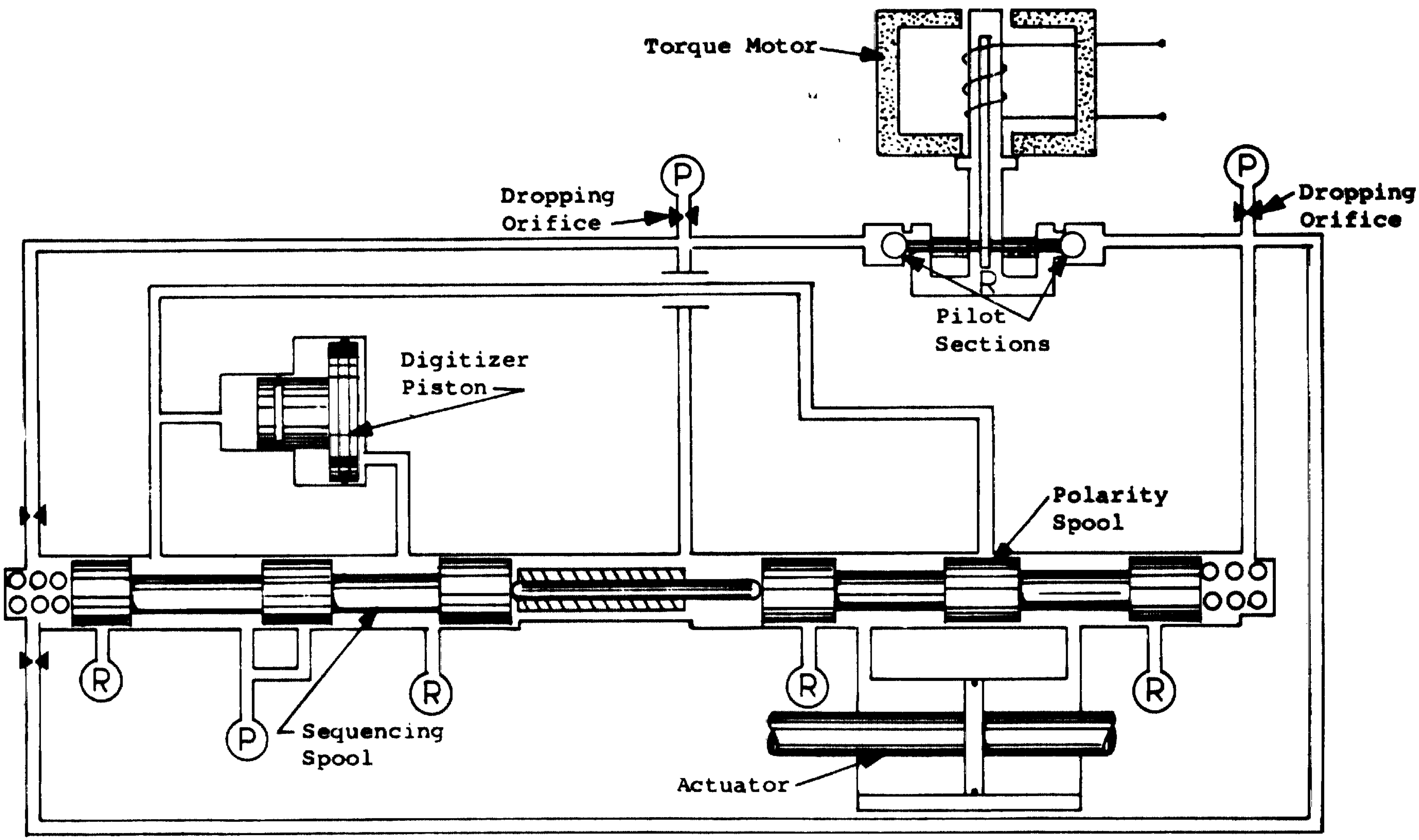

FIGURE A-15 SINGLE SHUTTLE SYSTEM

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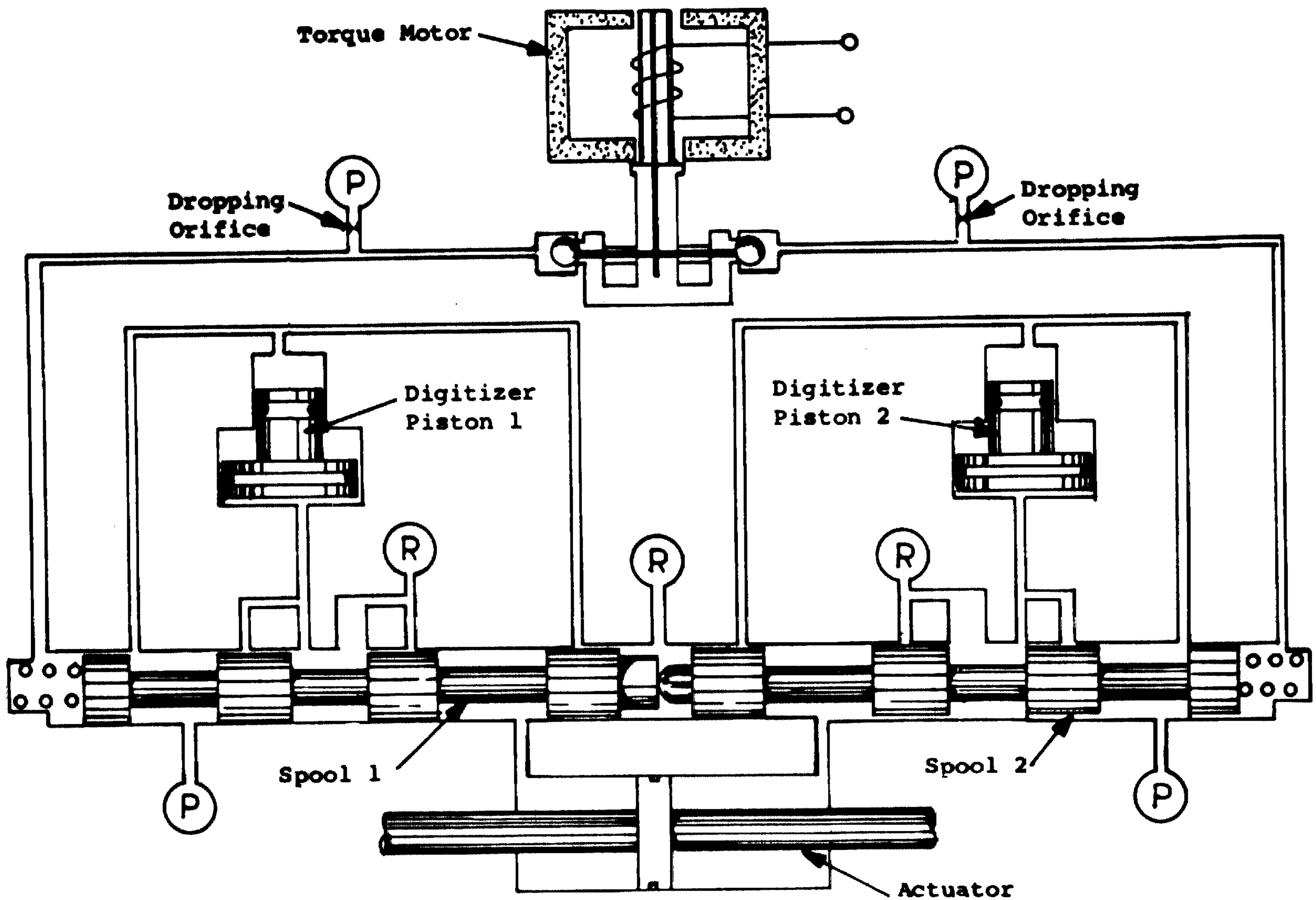

FIGURE A-16 DUAL DIGITIZER SYSTEM

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CONCEPT 10-049 BINARY INPUT, SOLENOID ACTUATED CAM DELAY UNIT

This approach, for a unit operating from a parallel binary input signal, is accomplished by use of several solenoid actuated cam delay units. One solenoid is provided for, and actuated by, each bit in the binary input signal. The cams are all driven from the actuator output through a differential gear train system.

A brief review of the principles of operation of the differential as used here will aid in understanding the succeeding explanation. Picture a differential gear system consisting of three independent shafts, input, output A and output B. If the input shaft is rotated one revolution, output shafts A and B will rotate one revolution each if they are not constrained. However, if the input shaft is rotated one revolution and output A is restrained from rotating, output B will rotate two revolutions. Likewise if the input is rotated one revolution and output B is constrained, output A will rotate two revolutions.

Now referring to Figure A-17 it can be seen how this differential characteristic is utilized. Shown in the figure is a simple system which operates from a binary system with only two bits. Each of the solenoid actuated cam delay units A and B operate in a manner similar to the incremental system cam delay unit. The only difference being that now the outputs from the pilot valves operate a pressure operated main flow control spool. As long as either of the units is energized the main spool will remain in the actuated position.

If a pulse is sent to unit A the solenoid will actuate its pilot spool. This will actuate the main spool causing actuator movement. The rack on the actuator shaft drives the differential input shaft through the pinion. Since unit B was not energized the B output from the differential is constrained. Thus there will be two turns of differential output A for each turn of the differential input. The 2:1 gear reduction reduces this back to one turn of the A cam for the two revolutions of the A differential output shaft.

Hence, it can be seen that when the differential input shaft has been driven one revolution, the A cam will also have made one revolution. At the end of this cam revolution the follower attached to the A pilot will drop back into the cam slot deactuating the A pilot's spool. The main spool will then deactuate, stopping the flow to the actuator.

In a similar manner, it can be seen that if B solenoid only is actuated, the differential input shaft will be driven two revolutions before pilot spool B is allowed to shut off. Now if both A and B solenoids are energized simultaneously the following action results. When the differential input shaft has been driven two turns differential output shaft A will have been driven two turns and cam A will have made its one revolution deactuating pilot spool A. Simultaneously, differential output shaft B will have made two turns and cam B will have made only one-half turn.

CONCEPT 10-049 - (continued)

With output shaft A now constrained, one turn of the differential input shaft will now cause two turns of the differential output shaft B and one-half turn of cam B. B has now made one complete rotation and pilot spool B will deactuate causing the main spool to deactuate also and stop the actuator movement. This principle can be carried to as many bits as desired.

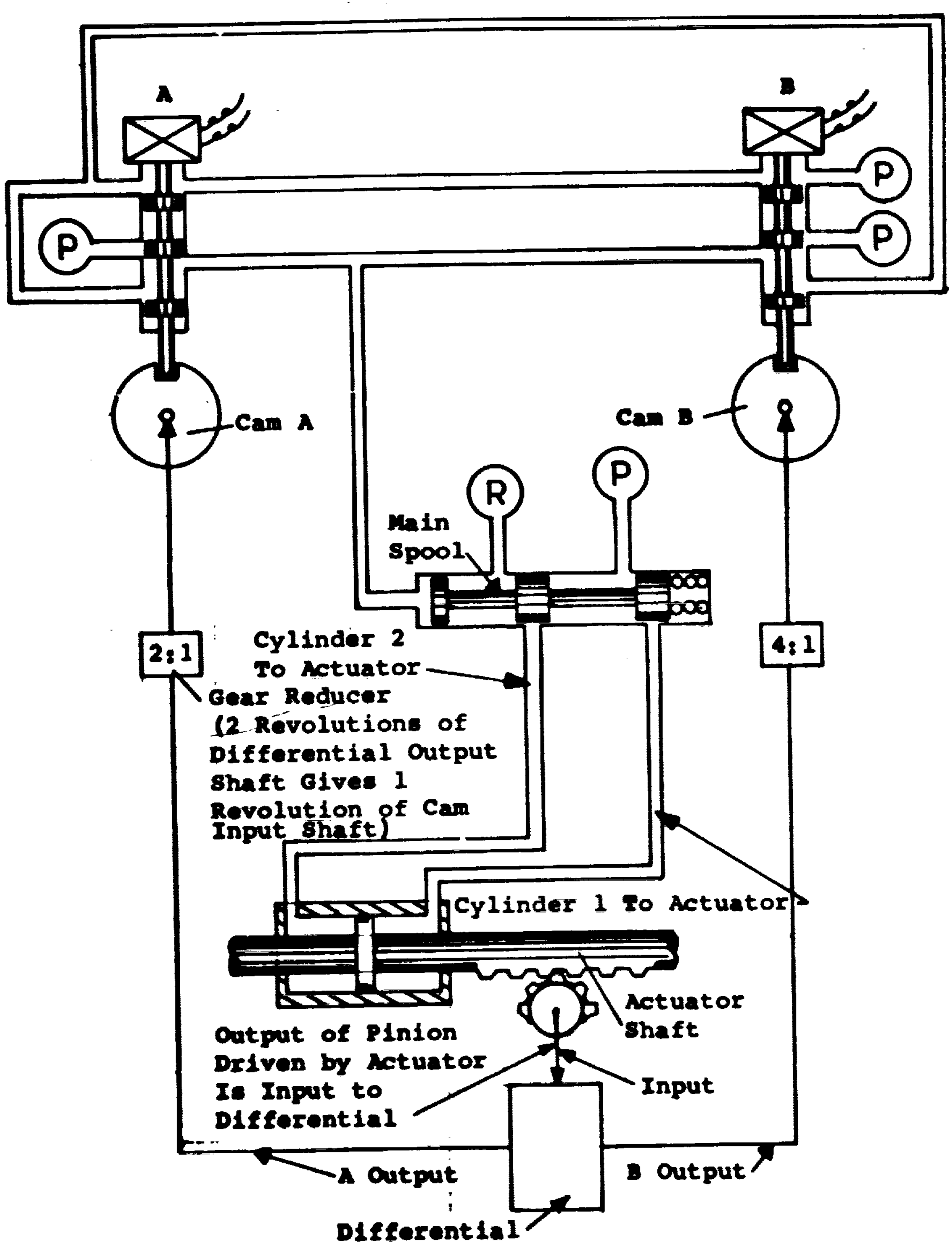

FIGURE A-17 GEAR CAM SCHEMATIC

CONCEPT 11-141 ELECTROHYDRAULIC LINEAR STEPPER ACTUATOR

The novelty of an Electrohydraulic Linear Stepper Actuator (E.L.S.A.) is in the fact that a relatively large number of discrete, accurate positions can be obtained with a small number of digital control valves.

E.L.S.A. is a hydraulic cylinder with a control valve having "transmitter" and "receiver" ports (Figure A-18).

Transmitter ports are associated to make pairs which are activated one after the other. For a given activated pair one port is connected to high pressure supply, the other to low pressure line.

Receiver ports are interconnected and are part of the movable element of the actuator.

The actual positioning of an activated transmitter port pair with a receiver port locks hydraulically the cylinder piston in a precise selected position. Following position of the actuator may be obtained by supply commutation of transmitter port pairs. This is simply obtained by a selector valve.

One can now understand that an E.L.S.A. fitted with a distributor provided with n transmitter ports and m receiver ports may select N = n m different accurate positions by action of the selector valve.

Advantages:

- Open loop operation
- Position accuracy determined by machined parts
- No drift
- Efficient hydraulically
- Discrete control signals
- Crude cheap valves

Disadvantages:

- Speed limited by valve time constant and step size
- Large number of parts

Typical Commercial Unit:

Differential Linear Actuator, Model AE 34 502

Number of transmitter ports	n = 4
Number of receiver ports	m = 26
Number of selected positions	N = n m = 104
Supply pressure	up to 5,000 psi
Load	280 daN (620 lbs)
Total travel	52 mm = 2.05 in.
Step length	0,5 mm = 0.02 in.
Max. speed	190 steps/second

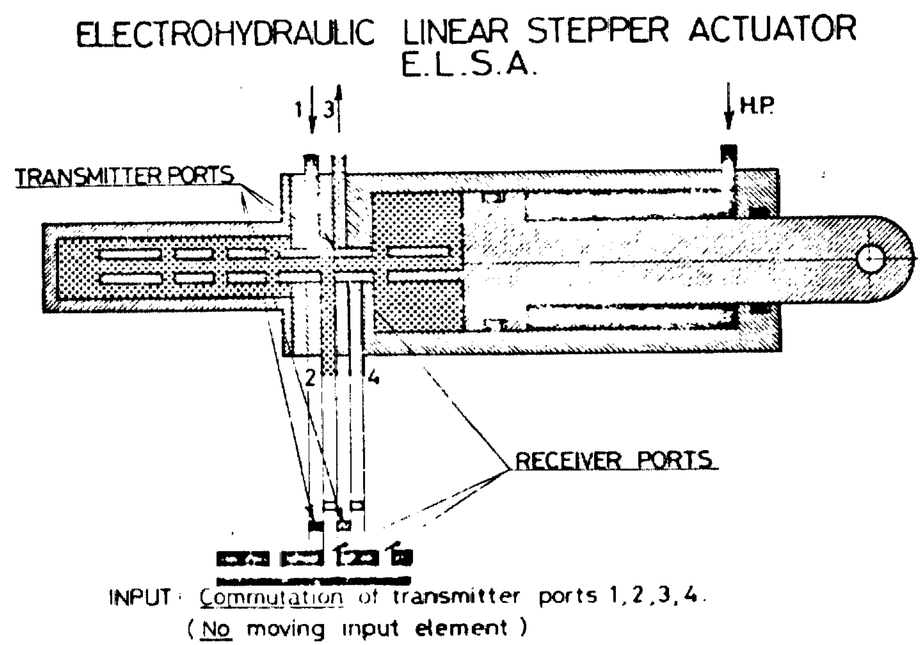

FIGURE A-18 ELECTROHYDRAULIC LINEAR STEPPER ACTUATOR (E.L.S.A.)

CONCEPT 12-073 "JO" BLOCK ACTUATOR

Cadillac Gage Company has developed a novel linear servo actuator that accepts parallel, binary digital control inputs and, without the use of a feedback device, positions an output as a function of these inputs.

The basic actuator, Figure A-19 consists of a housing with a main bore that contains four separate pistons. Each of these pistons has an outside diameter fitted to the main bore and two L-shaped extensions. An integral end cap and push rod are fastened to the housing at one end of the bore and a gland is fastened to the housing at the opposite end. The output member, which serves as a half area piston and connecting link, is retained by this gland.

In operation, system pressure is applied to the chamber formed by the end gland and the half area piston. This pressure exerts a biasing force on the piston assembly which tends to retract it. The chambers formed by adjacent pistons are supplied with system pressure or are vented to return by means of three-way, solenoid-operated valves. When system pressure is applied to one of these chambers, pressure is exerted on the opposing faces of two consecutive pistons, forcing them apart until the L-shaped extensions on the connecting rods and push rods in the chamber limit further motion. When the chamber is vented to return, the opposing faces of the pistons are forced together by the pressure on the half area piston until the push rod bottoms on the adjacent piston. The measured difference between these two piston positions is the stroke of the specific piston combination. Therefore, the length of the connecting rods and push rods determine the stroke.

The connecting rods and push rods are sized to produce binary positions as shown in Figure A-20.

In actual practice, the L-shaped sections are replaced by "T" shaped sections to remove moments applied to the individual pistons.

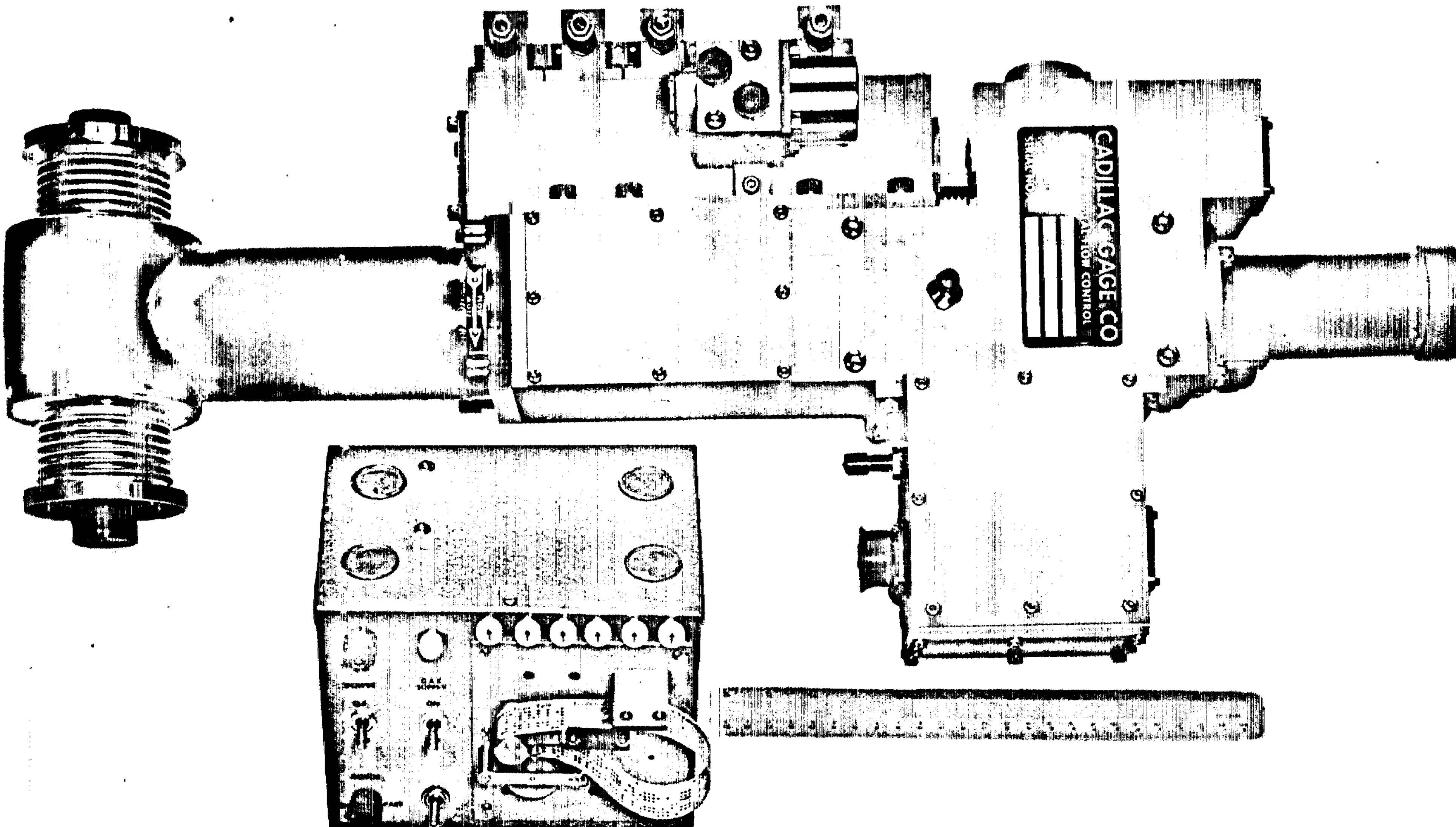

FIGURE A-19 DIGITAL FLOW CONTROL VALVE ASSEMBLY

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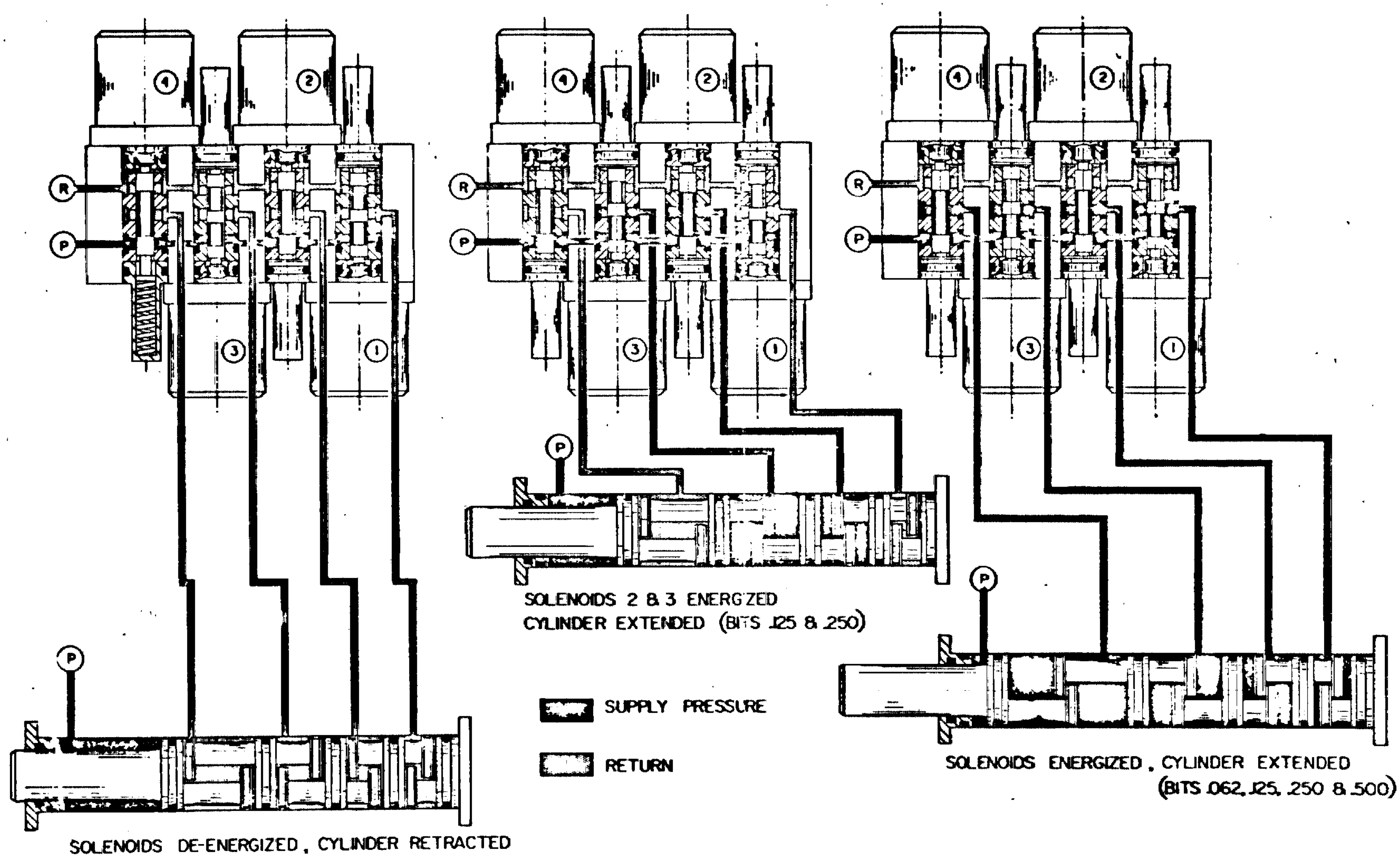

FIGURE A-20 DIGITAL FLOW CONTROL VALVE HYDRAULIC SCHEMATIC

CONCEPT 13-003 HYDRAULIC DIGITAL ACTUATOR

The idea of fluid-mechanical actuators that can work directly from digital input information has always been attractive. To eliminate the DA converters, servoamplifiers and servovalves, found in today's fluid-mechanical systems, and to be able to position an output shaft with the fine resolution and accurate repeatability of digital systems, have been goals worth attaining. The problem has been to build one.

Many designs - incremental pulse, serial absolute and parallel absolute inputs have been tried. It is an absolute actuator that accepts a parallel 8-bit straight binary coded electrical input to position an output shaft lineally. In its general form the actuator consists of a series of pistons with binary weighted displacements whose collective movement positions the output shaft.

The pistons are pressurized and depressurized through three-way, two-position, on-off valves; the amount each piston is allowed to move is determined by fixed mechanical stops machined when the actuator is made. The number of discrete positions that the output shaft of this type of actuator can assume equals 2n, where n is the number of pistons.

The prototype model, Figure A-21 has 10 pistons - 8 of them binary weighted active pistons and 2 passive pistons. Active pistons 1 through 8 determine the steady-state position of the actuator's output shaft; when the pistons are pressurized they move until they touch the mechanical stops. Passive pistons 9 and 10 act as buffers to dissipate the kinetic energy imparted to the load on full scale movements; one buffers forward motion, the other buffers reverse.

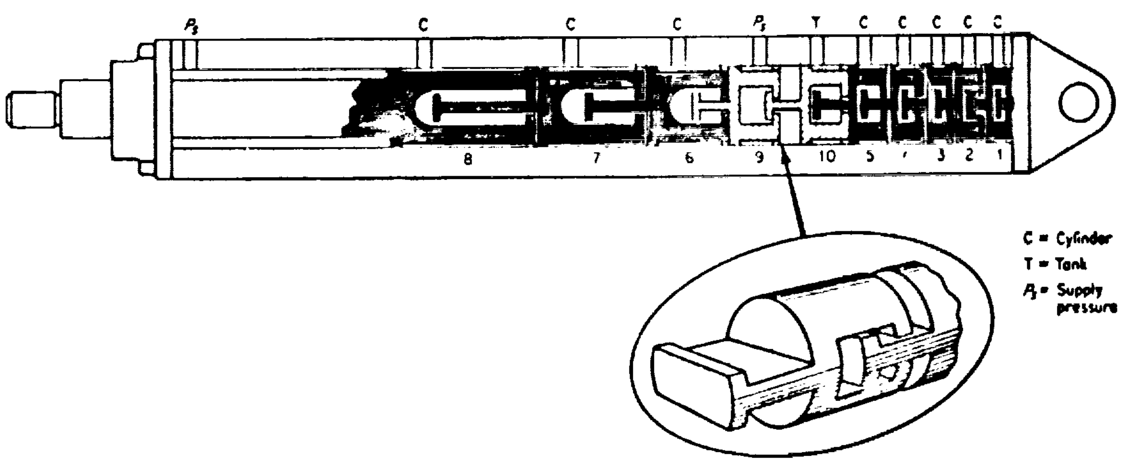

FIGURE A-21 Eight active and two passive (crosshatched) pistons convert binary-coded electrical input signals into lineal motion of the output shaft.

CONCEPT 13-003 (continued)

Because of the buffer pistons, the mechanical stops for the active pistons never have to bring the load to rest, so can be built lighter. But the buffering action allows the output shaft to overshoot the commanded position by a maximum of 1/32 in. on both forward and reverse full scale strokes.

The output shaft can take up 256 discrete positions in a full stroke distance of 2.55 in. The minimum movement is 0.010 in., called for by the least significant bit of the 8-bit binary input signal. Two versions of the three-way, two-position valves have been made. One, Figure A-22 , a miniature fast-response type, takes low electrical input power (0.12 watt), and its two-stage design needs hydraulic flow only when the valve is changing state. The other valve is integrated with the actuator housing. It has a slower response than the miniature type, is larger, and takes more input electrical power (28 watts).

The actuator with the miniature valves was developed primarily for aircraft and aerospace applications where the increasing use of on-board digital computers for navigation and guidance systems is creating interest in a direct digital link with the final control elements. The integrated design is for such industrial uses as operating process control valves.

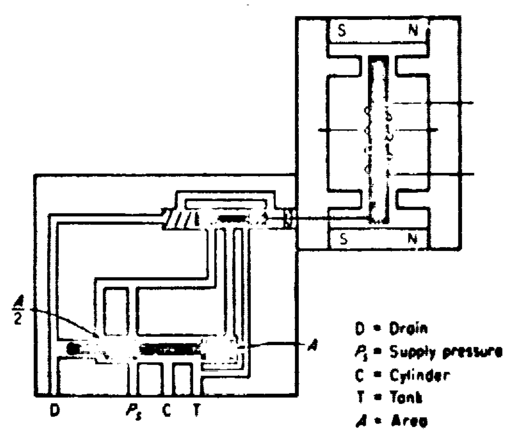

FIGURE A-22 Miniature on-off valve is part of actuator designed for aircraft and aerospace use.

CONCEPT 14-029   680J SECONDARY ACTUATOR

The secondary actuator is basically a fixed body quadruplex force summing, electro hydraulic servomechanism. It is a self-contained unit consisting of four modular independent servo controlled elements that are bolted to the front frame subassembly where the output quills of the individual elements are mechanically connected to a common output rocker arm. Each servo controlled element accepts individual electrical signals and is connected to individual high pressure hydraulic supplies. Each element contains the same components; i.e., servovalve, position feedback transducer, solenoid valve, differential pressure sensor and actuator cylinder. The four individual actuator rams are tied together to form a common force summing rocker arm output as indicated in Figure A-23. Each element requires only one hydraulic supply, thus maintaining complete physical separation of hydraulic systems.

The actuator is capable of two-fail-operate performance. Each actuator element incorporates a differential pressure sensor that produces a signal used to disengage a faulty element through pressure cut-off by the shutoff valve in each element. Three variations of the basic actuator are utilized for the 680J program. The output arm and centering requirements are the only variables. The left lateral and/or directional and the right lateral secondary actuators are designed to be returned to neutral by centering springs, and hold after three similar failures. The longitudinal secondary actuator contains a brake to hold in its failed position after three similar failures. See Figure A-23.

The hydraulic schematic of the secondary actuator is shown in Figure A-24. The hydraulic circuit of only one element is presented in the schematic since all four of the elements are hydraulically identical. The engagement and shutdown of each element is controlled by the two-way, shutoff valve which supplies pressure to the disengaging piston and servovalve. When the shutoff valve in any one module is turned on, supply pressure is applied to the disengaging piston which pushes on a common bar in the centering mechanism which releases the actuator output. A single element may be engaged since each piston produces sufficient force to retract the locking spring. Each element has ports for the connection of a hydraulic oil supply (pressure and return) and its own body mounted electrical connector. Each module is therefore totally hydraulically and electrically isolated from the others. Each module is a closed loop actuator since the module contains an electro hydraulic servovalve, a cylinder and output ram and an LVDT for electrical position feedback. Two additional components, related to failure correction, are the on-off solenoid valve and the pressure sensor with its LVDT for producing an electrical signal for failure indication.

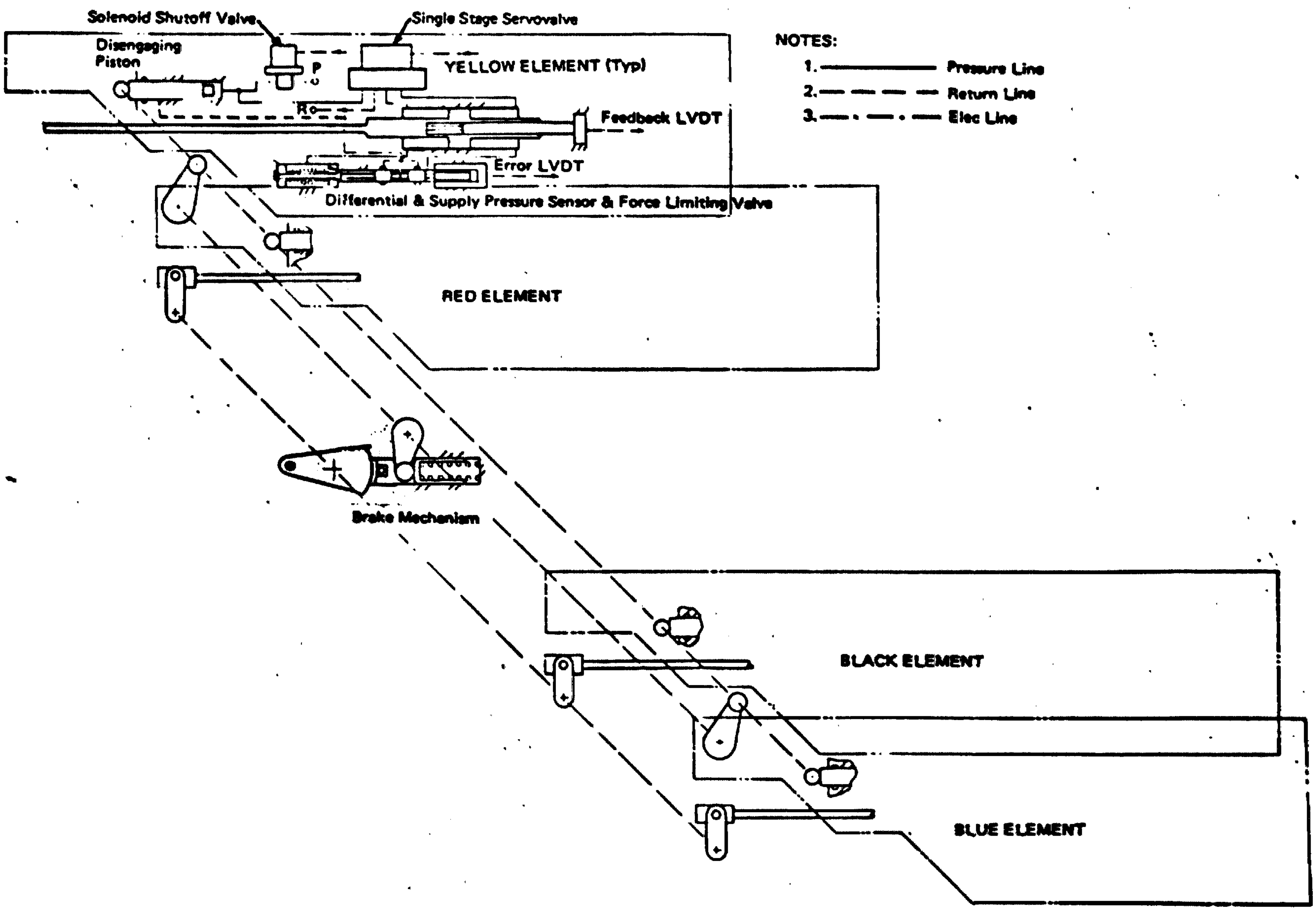

FIGURE A-23 SECONDARY ACTUATOR HYDRAULIC MECHANICAL SCHEMATIC

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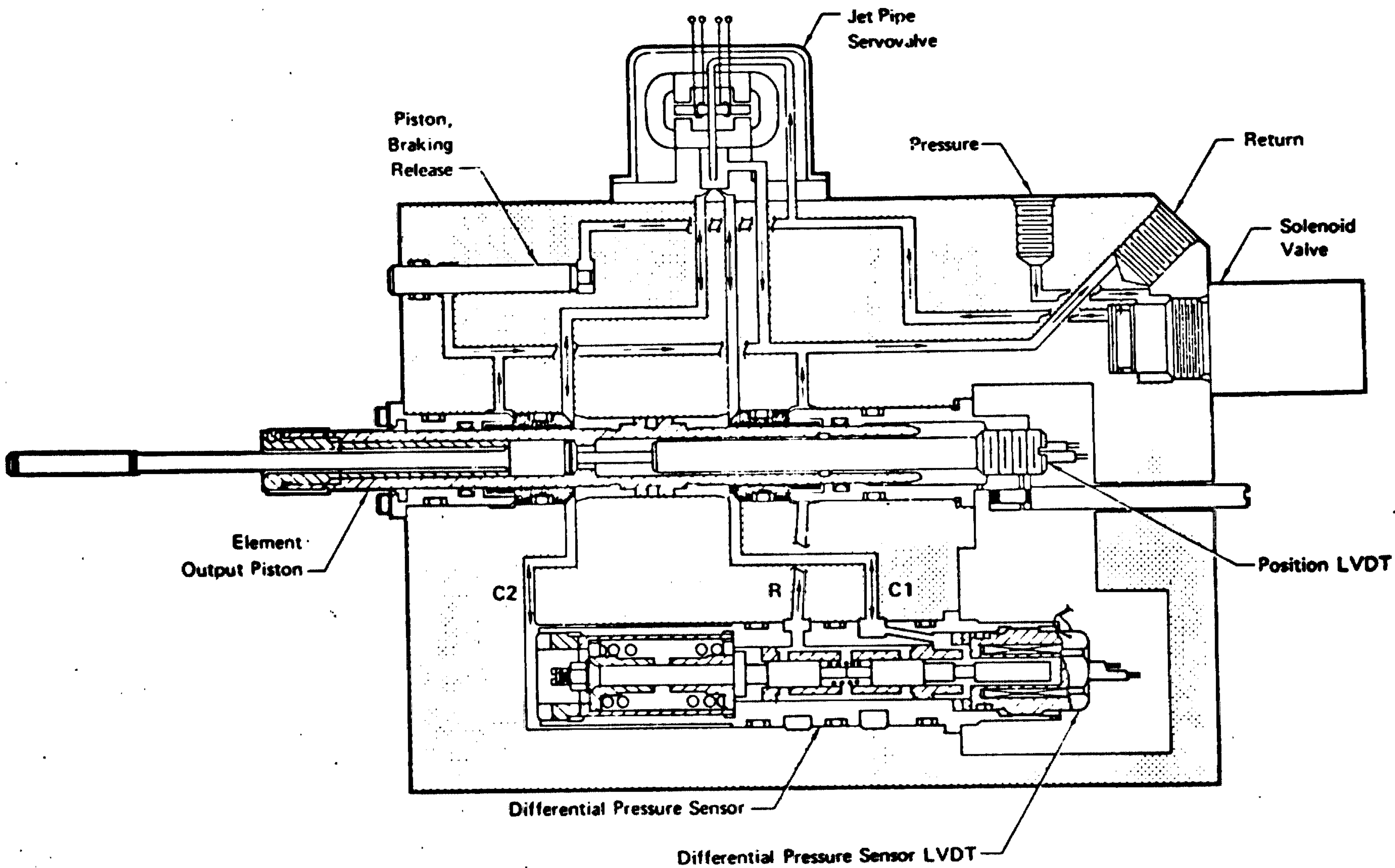

FIGURE A-24 HYDRAULIC SCHEMATIC, SINGLE ACTUATOR ELEMENT

CONCEPT 15-106 4-CHANNEL INPUT ACTUATOR

Working jointly with Honeywell, National Water Lift built a 4-channel input servo actuator demonstrator for fly-by-wire applications that provides two-fail operation with minimum failure transients. Please refer to the attached schematic of one typical channel, Figure A-25.

The actuator uses four identical servo modules linked to a common force-summing shaft. Each line replaceable servo module is independently driven by one channel of the FCS and is hydraulically isolated from adjacent modules. Hydraulic equilization provisions are incorporated in each module. A key performance feature of the actuator is force-summing which greatly minimizes the effects of failure and switching transients. Only minor performance degradation occurs with a channel servo failure since the three remaining operable channel servos continue to oppose the failed servo until it is disengaged from the system.

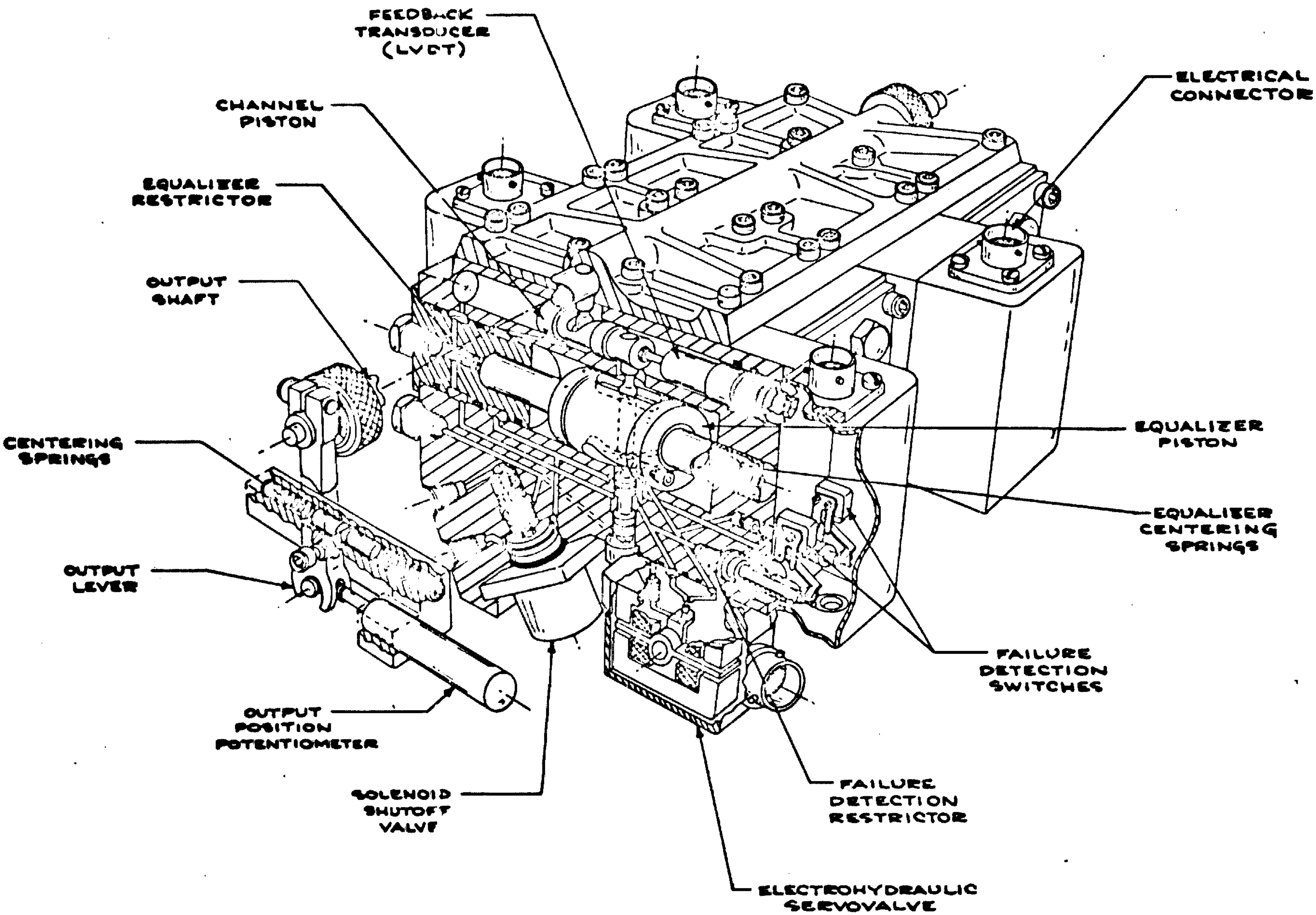

FIGURE A-25 4 CHANNEL INPUT ACTUATOR

CONCEPT 16-016 TRIPLE REDUNDANT ACTUATOR

The three systems can be operated individually or simultaneously. Unless one of the hydraulic supply pressures is lost, there is no loss in output force or stiffness when changing from multiple to single system operation; the servoram force summation results in simultaneous operation of the three output rams regardless of how many channels are driving. A signal from the servo amplifier to the servovalve causes fluid flow to the servoram, as shown in Figure A-26. Position feedback from the servoram is provided by rotary variable differential transformers on the force-summing shaft driven by the servorams. The feedback signal sums with the command signal to the amplifier, and the servoram keeps moving until the sum of the signals is equivalent to the servovalve current necessary to balance the spring load on the servoram.

Unlike a normal single servomechanism, the load on the servoram also includes the possibly unequal force outputs from the other servorams. The resulting force "fight", a tendency toward cancellation among the three systems, is reduced by pressure equalization. Differential pressure transducers measure the output of the individual channels. These signals are voted in a mid-value selector and an error signal is generated to force each channel to approach the midvalue. The error signal is a measure of the "goodness" of a channel and hence provides the actuator monitoring point.

An idler link, attached to the force-summing shaft crank, leads to a summing link where the servoram output position and a mechanical input are added and feed into the main control valve lever (Figure A-27). Flow from the main control valve causes motion in the output ram in the direction to cause the feedback lever to close the valve. The mechanical input is not necessary to the operation of the actuator, and is provided only as an additional safety feature for initial flight testing. The mechanical linkage can be disconnected and the link tied to the structure for true fly-by-wire operation.

The feasibility and practicality of the fly-by-wire redundant actuation system were demonstrated. Design features and performance capabilities necessary to meet the reliability, performance, and safety requirements of a flyable system were incorporated into the actuation system. Although the equipment was intended for laboratory investigations, it was designed to meet airworthiness specifications and can be flight tested following limited qualification testing. The actuation system is unique in that fewer moving parts are contained in the triplex forcing mechanism than in other two-fail-operational actuators. Also, the inherent failure transient suppression characteristics provide for minimum system failure transients.

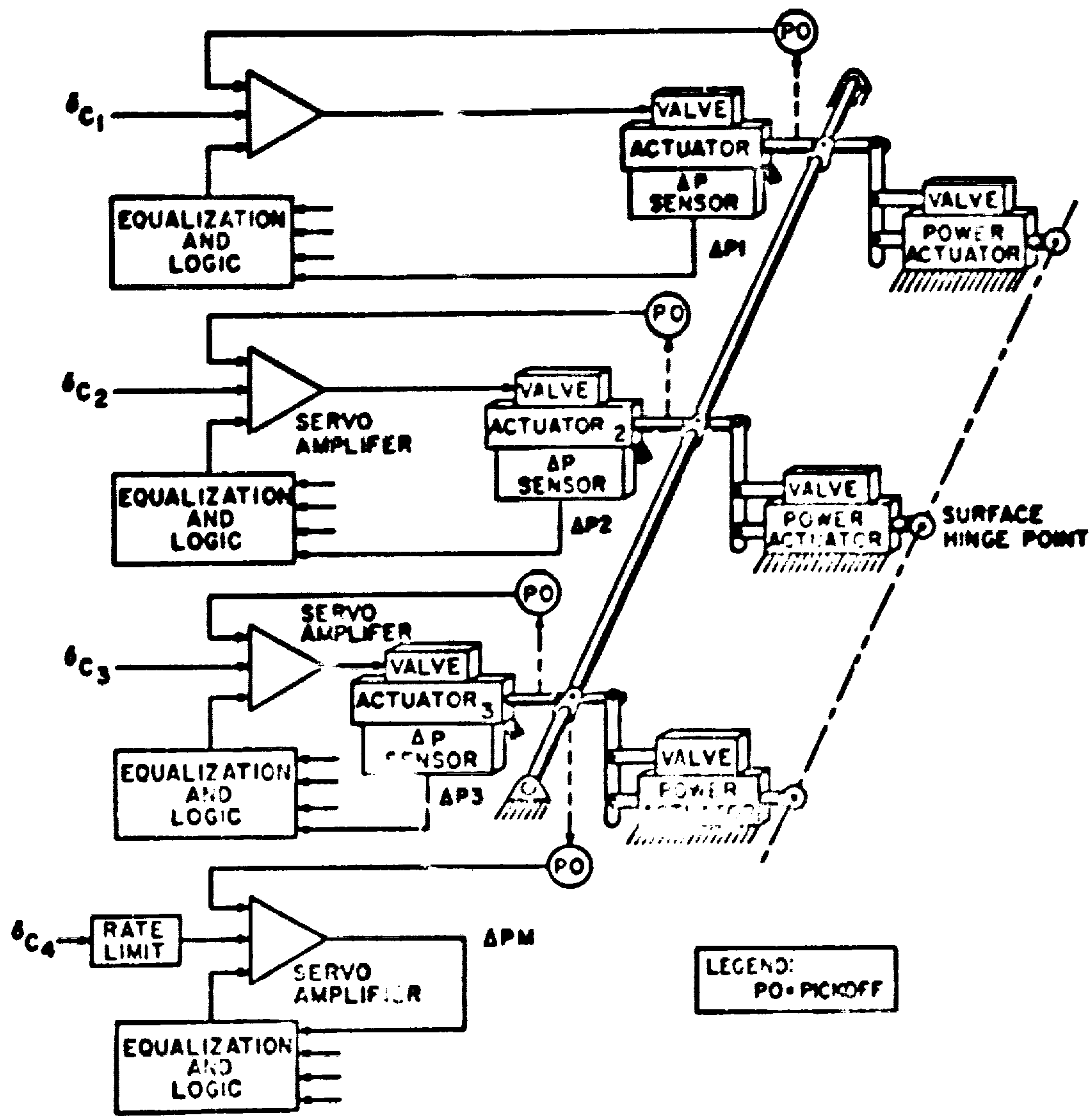

FIGURE A-26 TRIPLE REDUNDANT ACTUATOR SYSTEM DIAGRAM

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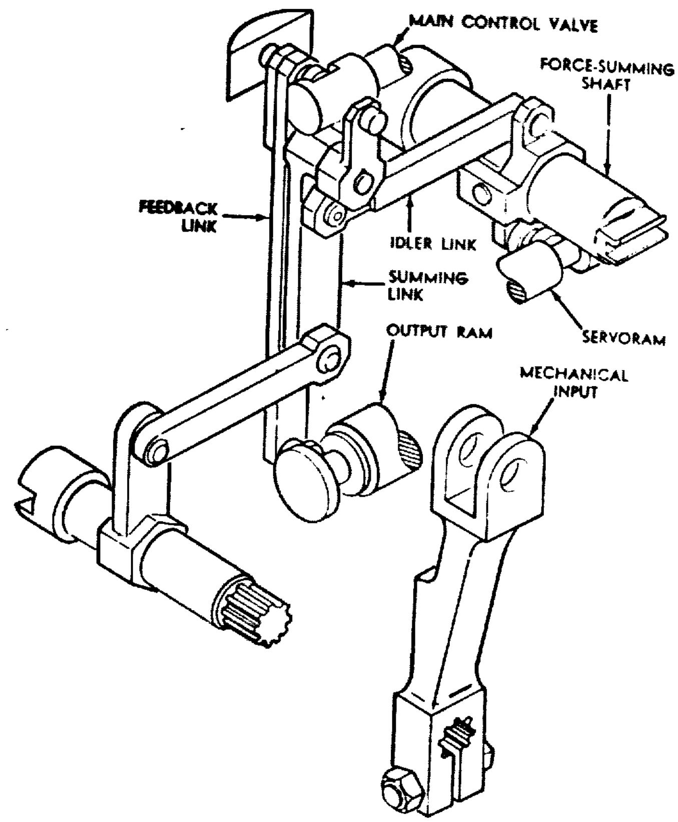

FIGURE A-27 TRIPLE REDUNDANT ACTUATOR ASSEMBLY

CONCEPT 17-002 TRIPLE REDUNDANT AC SERVOACTUATOR

The control system is shown in Figure A-28. The electrical portion operates from an AM, 400-cps carrier supply. The motion of the control stick is converted into three identical electrical signals by three power-type, linear, variable, differential transformers (LVDT). Each electrical signal is transmitted over separate electric cables to the input of its associated servovalve. The autopilot input is also an AM carrier and consists of three identical signals transmitted over separate cables to a switching point for selection of manual or autopilot control.

Each of the three valves is an ac input, two-stage, four-way, jet-pipe servovalve with internal hydraulic feedback and a mechanical input. The three parallel control channels are interconnected by mechanical ganging of the three output spools (second-stage spools) and also by using a common piston rod for the three actuators, as shown in Figure A-29. The common actuator rod positions the load. System feedback is provided by a feedback rod that produces an identical mechanical input to each servovalve.

The electrical input signal is converted into a developed mechanical torque on the armature. This torque is applied to an armature spring that supports the armature and constrains its rotation. The jet pipe is connected to the armature and discharges into two control chambers as shown in Figures A-29 and A-30 Any displacement of the jet towards one chamber will increase its control pressure and reduce the pressure in the other chamber.

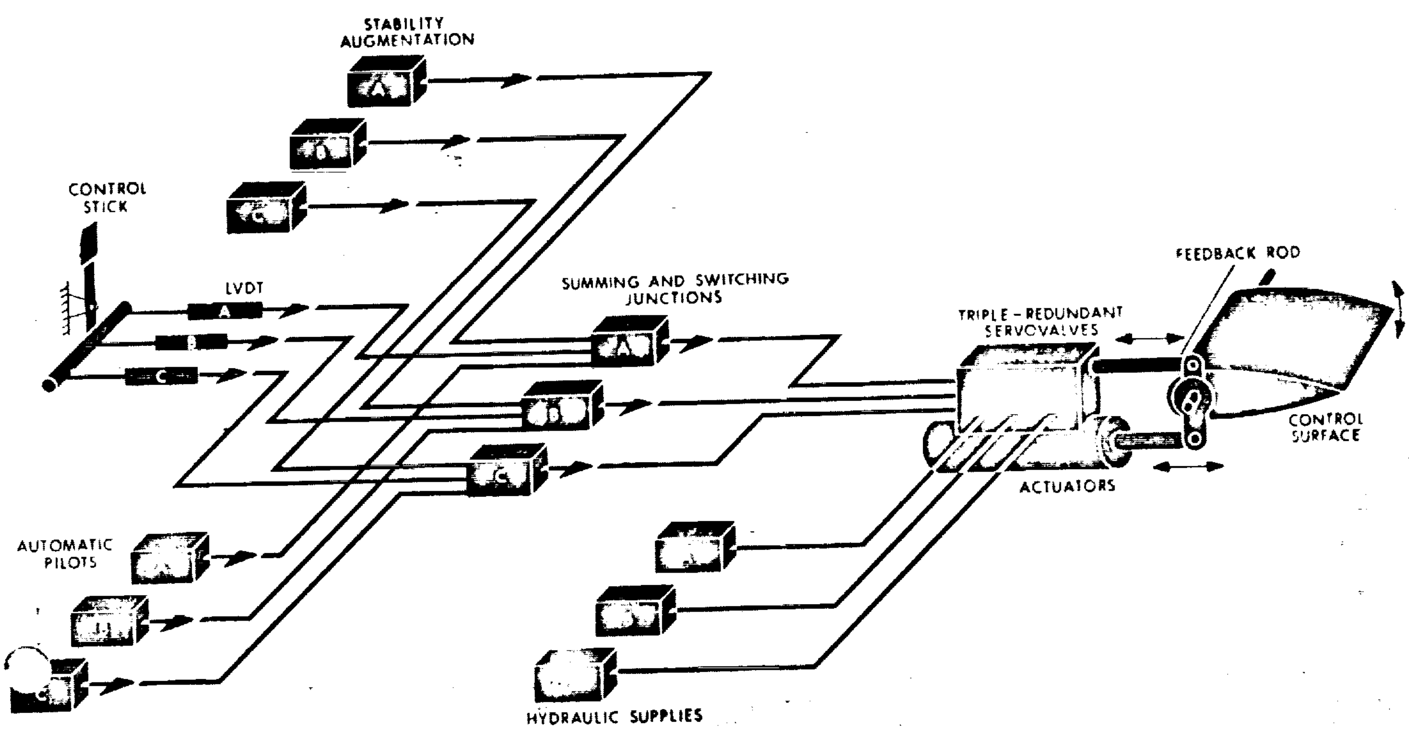

FIGURE A-28 A complete triple-redundant control system is illustrated. The three electrical and hydraulic channels are identical and completely isolated except for a common stick input and a common output rod from the tandem actuators. Within the control loop, all components and transmissions are for amplitude modulated, 400 cps carrier signals. This control system is capable of accepting dual inputs: (1) stick inputs from the pilot

FIGURE A-28 (continued) and (2) combined automatic pilot and stability augmentation signals. Electrical wires are used in place of the conventional complex mechanical linkages between control stick and the widely separated hydraulic control units.

The control pressures of one channel are transmitted to the opposite ends of its associated output spool, and any differential control pressure will generate a force to make the spool follow the jet pipe until the jet is equally positioned between the control chambers. This action produces the hydraulic feedback feature.

Displacement of any spool from its central position to one side will connect one of its output ports to the supply pressure and the other port to the tank. Each spool output is ported to its associated balanced, double-acting actuator, and any differential output pressure will generate a force on the piston rod which is transmitted to the load. Since the output spools of the three servo-valves are mechanically linked to form a common second-stage spool (or power slide) and the three actuators are also mechanically linked to form a common output actuator, the output represents the sum of the three individual channel outputs.

The position of the load is fed back to each servovalve torque motor through the feedback rod. The summing junction of the input and feedback signal is the armature spring. The torque developed by the electrical input signal acts against the armature's torsion spring. The platform of the spring end opposite the armature end is not fixed relative to the valve's body, but can be rotated in proportion to the displacement of the feedback rod. The rotation of the spring platform, in response to load motion initiated by an electrical input, will continue until the platform has rotated through an equal but opposite angle to that of the armature end of the spring. This places the armature and jet pipe in the central position and reduces the differential control and output pressures to zero, so that there is no force to move the load.

An electrohydraulic flight control system, such as the triple-redundant, a-c servovalve, offers many advantages over the standard, mechanical-hydraulic equipment used extensively today. The electrohydraulic system has high response, light weight, and is easy to integrate into automatic-flight and stability-augmentation systems. The parallel redundant design insures high reliability.

With each passing year, aircraft and spacecraft requirements become more and more demanding. It is in these high-performance vehicles that electrohydraulic flight control systems similar to the one described in this paper will be used.

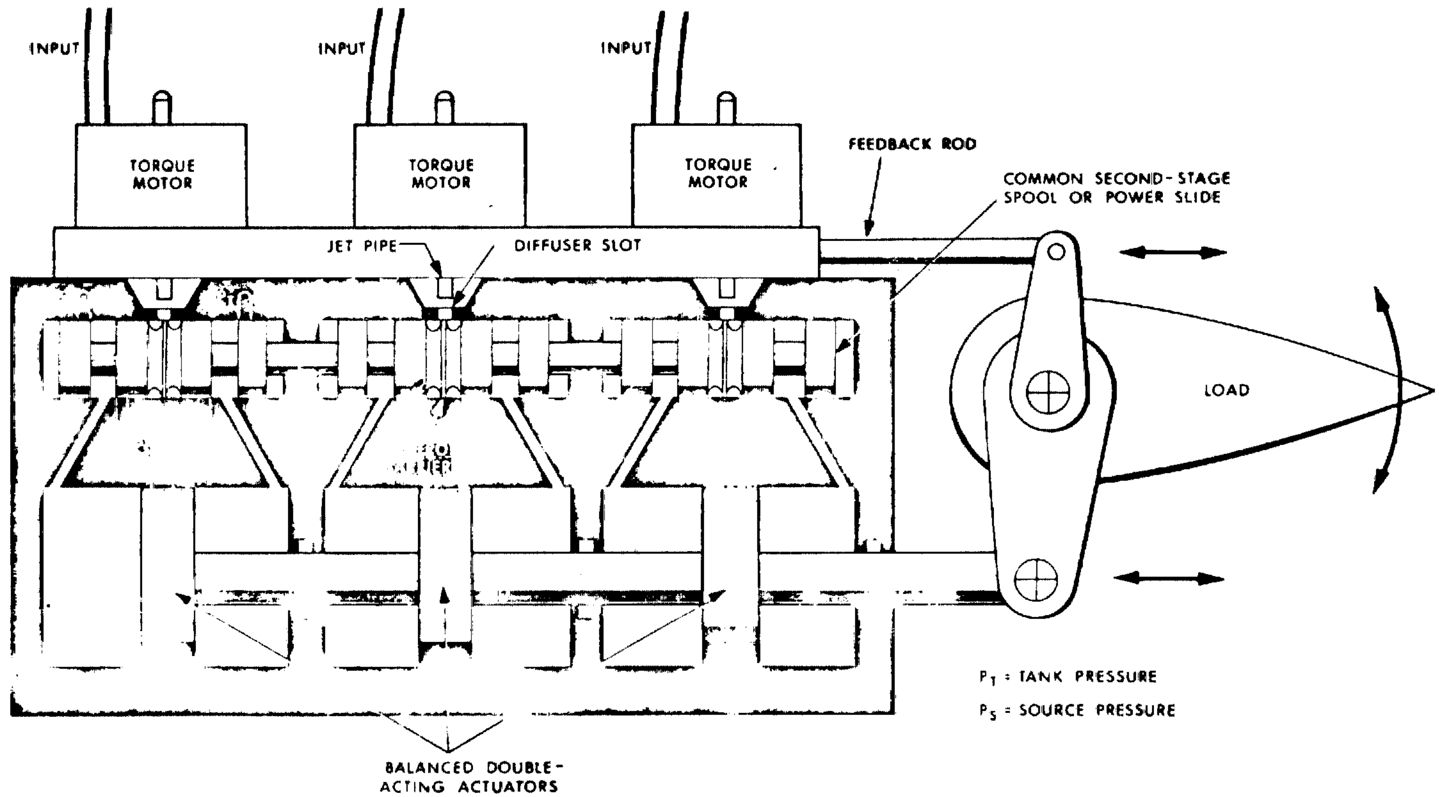

FIGURE A-29 SERVOVALVE AND ACTUATORS

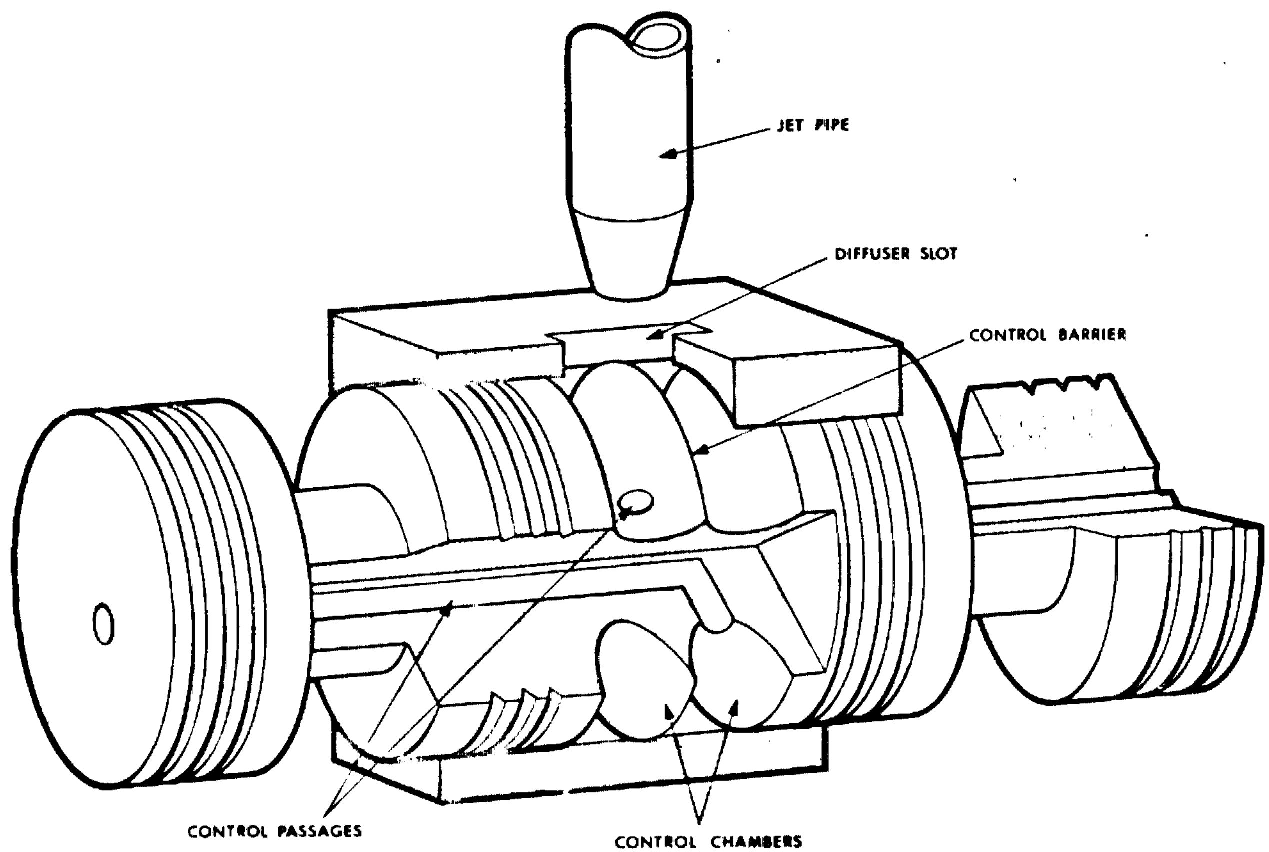

FIGURE A-30 JET PIPE DIFFUSER SLOT SERVOVALVE

CONCEPT 18-005 AD SKYRAIDER PITCH CONTROL SYSTEM

The system operates directly from ship's ac power to eliminate any dc conversion equipment. Therefore, the control stick position transducers are LVDT's (Linear Variable Differential Transformer), signal summation uses transformers, and the hydraulic servovalves use ac torquers. Figure A-32 shows a diagram of the system for the pitch axis. The system employs triple redundancy to obtain the desired reliability which is equated to the Douglas AD Skyraider pitch control system reliability. Monitoring is performed at the servovalve torquer which also serves as the summing junction for the servo input and mechanical feedback.

While the Douglas study showed that a fly-by-wire system could be designed without electronics or switching to match the reliability of a mechanical system, the study and the design had a number of failings. First, the study failed to include any discussion of artificial feel implementation which is vitally important to a practical fly-by-wire system.

Second, the ac servovalve torquers are very inefficient devices which require a great deal of electrical power from the stick position LVDT for operation, particularly since additional torque is required to operate with the mechanical feedback. The three valves require a total power of 50 watts. The triplex LVDT absorbs another 60 watts at its maximum displacement.

Third, the size and weight of the components are extremely high thus partially negating one of the basic advantages of fly-by-wire of size and weight reduction. The breadboard models of LVDT and servovalve (excluding the actuator) weigh 30 and 55 pounds respectively. Although flightworthy components would certainly weigh much less than this, the trend is obvious. For comparison, a triplex signal LVDT would weigh about 5 ounces.

Fourth, the magnetic summing and monitoring techniques are not practical for two reasons: (1) signals from different power supplies cannot be summed inductively unless they are exactly synchronized; otherwise the output signal will bear no significant relationship to the desired signal; and (2) because the transfer impedance of a transformer depends on the flux level in the core, the output level for one input signal depends on the presence and level of a second input. This nonlinear effect causes a varying forward path gain in the control system.

Fifth, the gradient of surface deflection per control stick displacement is reduced by one-third for each electrical channel failure. One channel failure reduces the command torque at the servovalve input to two-thirds normal which is balanced by the feedback torque produced by two-thirds normal surface deflection. The change in control authority would reduce system performance significantly even for the first failure.

Finally, the use of mechanical feedback and coupled servovalves presents very difficult design and synchronization problems. At lease 2 years were spent in developing a prototype model with only limited success. We conclude from the above evaluation that the Douglas approach is not suitable

CONCEPT 18-005 - (continued)

for use in fly-by-wire systems. Although the results were negative, the program has provided a beneficial contribution to fly-by-wire development because it will prevent others from attempting the same approach. Work for the Air Force by Douglas is still continuing but with redirection to include electronics and a different actuator approach.

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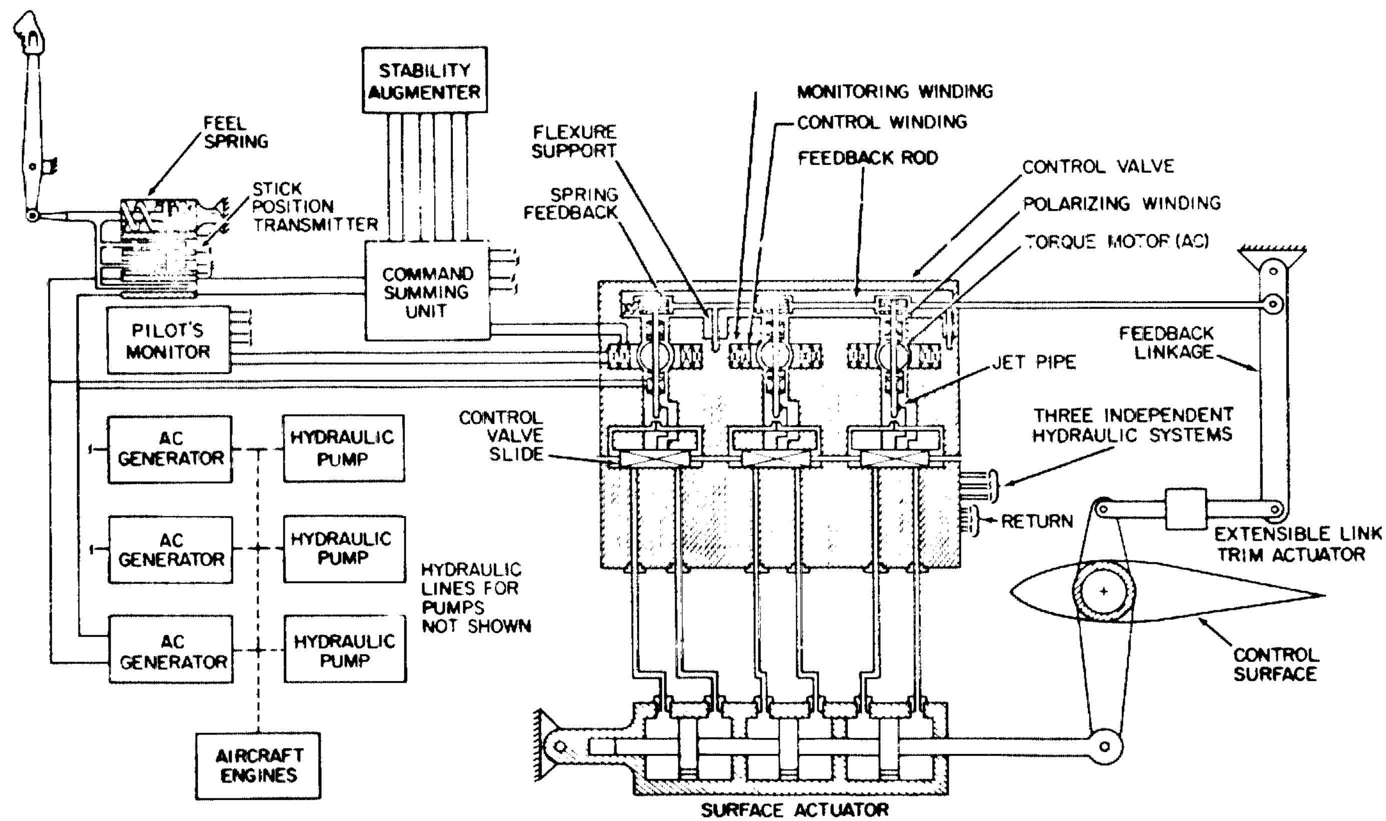

FIGURE A-31 AD SKYRAIDER PITCH CONTROL SYSTEM

CONCEPT 19-005 DUAL ACTUATOR SERVO SYSTEM

Fail-Passive Secondary Actuator configuration (Figures A-32 and A-33) is described in somewhat more detail because of its unique characteristics. It employs a small redundant secondary actuator which mechanically drives the main control valve and power actuator with nearly unity feedback. The dual mechanical linkage can be sealed within the actuator body where it is protected and bathed in oil. Both the secondary and power actuators employ active redundancy. When dual hydraulic supplies are used, the secondary actuator is dual tandem with two single-stage jet-pipe valves driving each piston thus forming four inner servo loops. When triple hydraulic supplies are used, the secondary actuator is triple tandem with a single valve driving each piston thus forming three inner loops.

The uniqueness of the configuration derives from the inner loops which are designed to have passive failure characteristics. A fail-passive channel fails in such a way that it has no output and it does not interface with the normal operation of a parallel channel. In other words, active or hardover failures have been eliminated by design. Since a failed channel has no force output, the other good channels can operate unimpeded. The single-stage jet-pipe valve not only has the proper failure characteristics, but it also acts like a very open-centered valve so that fluid can be forced back through it with relative ease thus preventing hydraulic lock. The servo error signal is formed in the position feedback transducer, rather than in an amplifier as is normally done, such that a transducer failure blocks the command signal. This feature prevents the open loop condition that normally results from a loss of feedback. The electronics fail passively because ac signals are used. A hardover electronic failure causes a dc output to which the ac circuits are not sensitive.

If a hardover input should occur in a channel or as an input, the other channels collectively offset the output force of the failed channel at the force-summing actuators. The high loop gains reduce the resulting position offset to an insignificant level. Therefore, a quadruplex servo with four hydraulic and electrical supplies will operate after three failures, and a triplex servo will operate after two failures. Further, no monitoring, switching, or engage valves are required in this approach. Monitoring is performed, however, primarily for failure reporting. In the triplex servo, a hardover monitor may be used to provide center and lock, in the event that one of the three failures is not passive.

Both the fail-passive triplex and quadruplex servos have been tested in the laboratory to demonstrate their operation and performance. The quadruplex servo operates slightly better than predicted by theory. This is because a failed servo does not completely bypass the other channel on the same piston. Therefore, after three failures, the actuator retains about 20 percent of its dynamic performance which would likely be enough to let the pilot fly the aircraft. An important point to note is that the servo operates after three failures without the use of monitoring or switching. The lack of switching not only simplifies the design but it also eliminates the failure transient problem. Relatively simple monitoring utilizes differential pressure transducers to measure the output force of the servos. The monitor correlates this information with the error signals to determine which channels have failed. A final point is that the fail-passive servo does not require tight tolerances because it need not be monitored. On the experimental model, the tolerances were purposely varied by $\pm$ 30 percent with no noticeable effect on performance. This result has obvious advantages in the economy of construction and operation.

The configuration has the following advantages:

(1) No switching required for failures thus eliminating switching transients and engage valves.

(2) Relatively simple monitoring required for failure reporting only.

(3) A triplex system remains operational after two failures; a quadruplex system remains operational after three failures; etc.

(4) Size, weight, complexity and cost are minimum for the given degree of redundancy.

(5) Very tolerant to channel mismatches.

(6) Easily adapted to any degree of redundancy.

(7) Requires single-stage valves rather than two-stage valves which improves reliability.

(8) Very tolerant to dirty fluid; can operate with 200 micron filters.

The configuration has the following disadvantages:

(1) Requires a secondary actuator.

(2) May have limited dynamic performance and threshold in very high performance applications when using presently available single-stage jet-pipe valves.

(3) The triplex configuration requires an electronic model to ensure center and lock for a third failure.

(4) Force degradation for hydraulic failures.

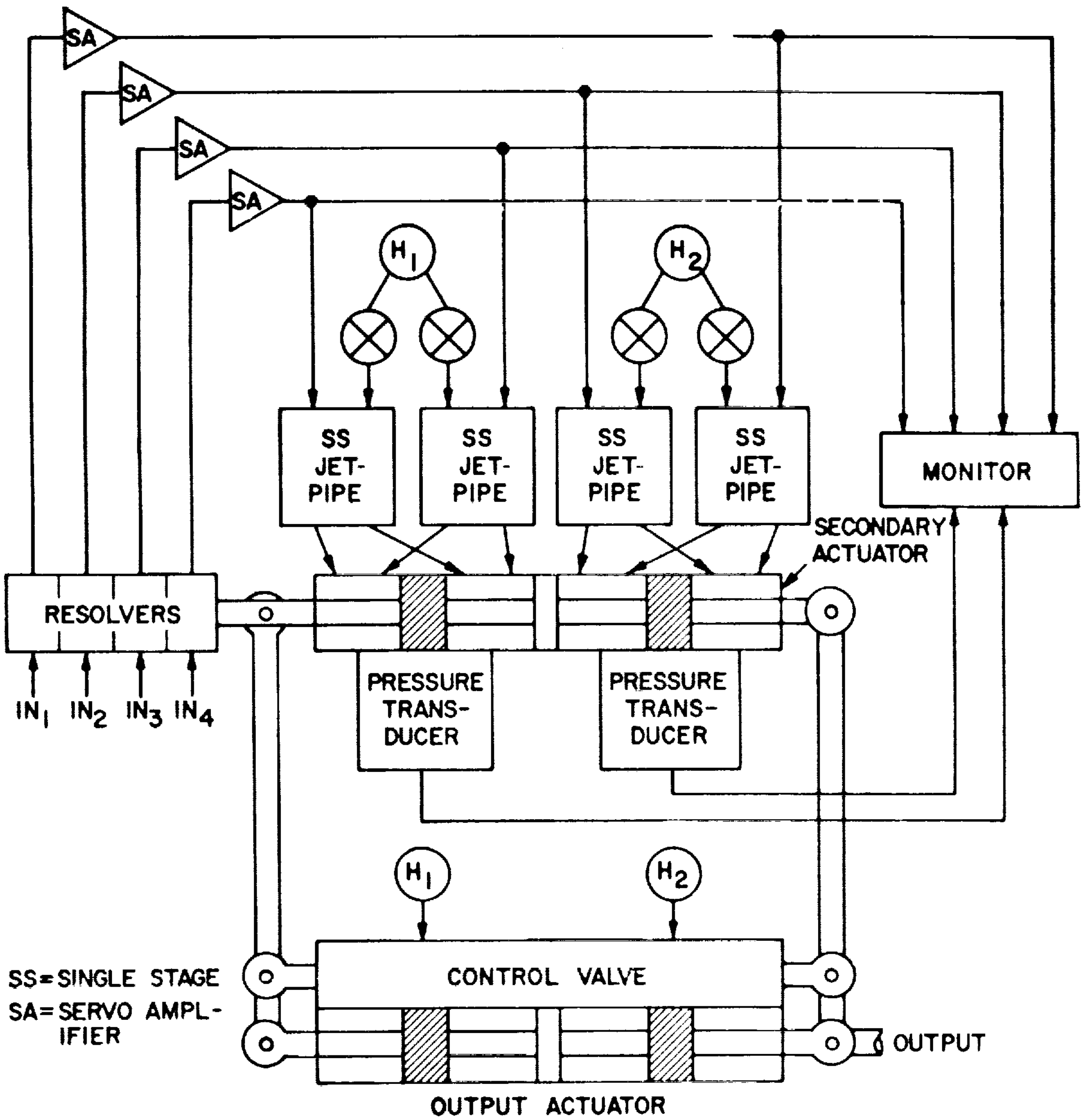

FIGURE A-32 FAIL PASSIVE SECONDARY QUAD LOOP ACTUATOR

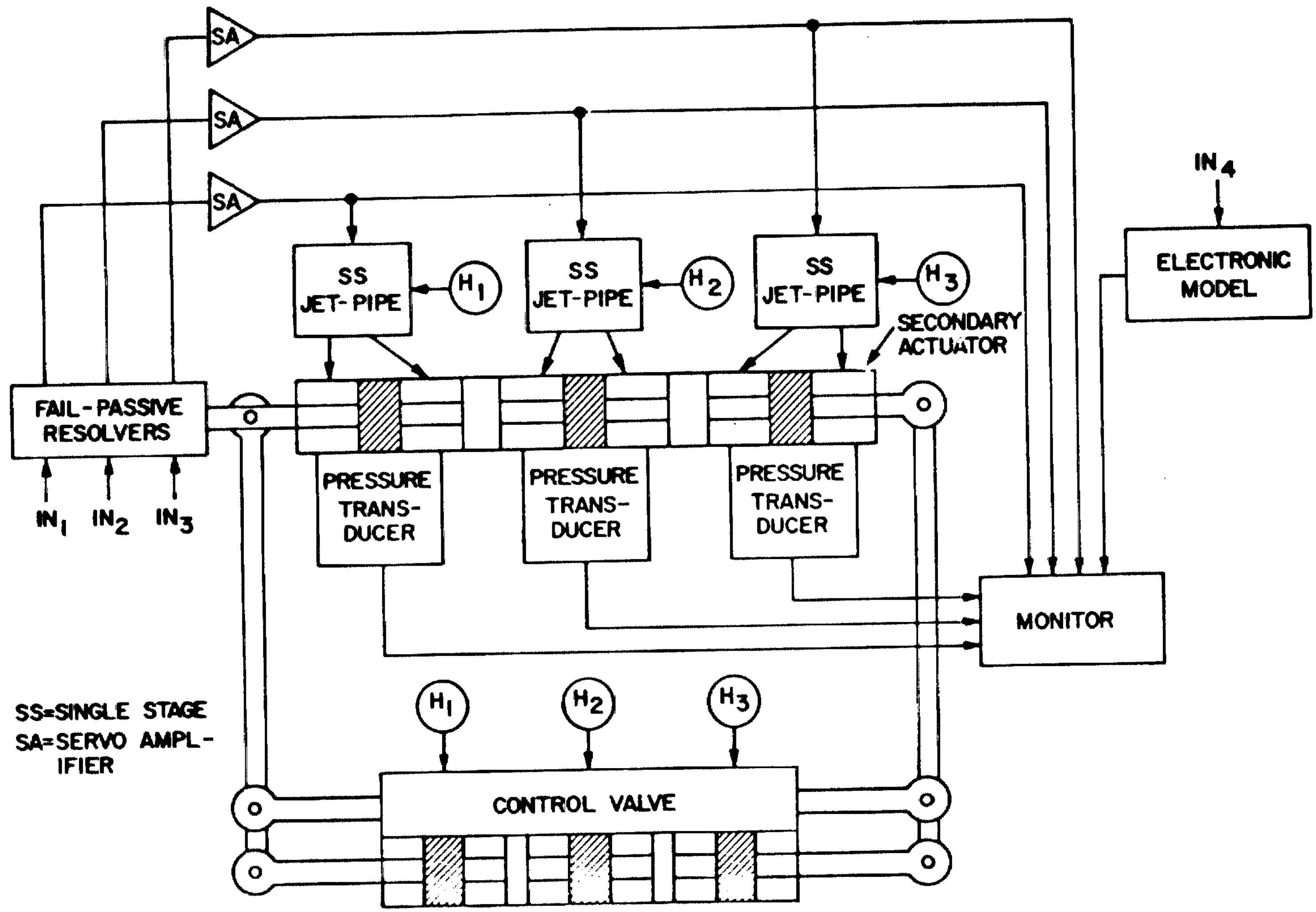

FIGURE A-33 FAIL PASSIVE SECONDARY TRIPLE LOOP ACTUATOR

CONCEPT 20-005 QUAD PARALLEL ACTUATOR SYSTEM

The Force-Summed Voted Actuator configuration, shown in Figure A-34 has been developed and tested by Elliott Brothers of England. It employs four separately controlled hydraulic actuators coupled in parallel to a common output member by means of miniature hydraulic couplings combined with ball clutches. Each hydraulic coupling has a zero rate springbox characteristic, and its stroke is determined by the tolerance between channels necessary to allow for component variations. If failure causes the coupling to reach the end of its stroke, the balls disengage from a groove in the common member and so declutch the failed actuator from the common output. The clutch mechanism is a variation of the well-known quick-release self-sealing hydraulic coupling which is in widespread use.

A simple gate mechanism is provided to prevent more than two channels from becoming disengaged at any one time. This gate is required to prevent disengagement of more than two channels which might otherwise occur due to some remote common cause, such as an excessive output load on the actuator, and cause a loss of control. The pilot can be warned of a declutched actuator channel by means of a failure display panel. The actuator remains disconnected from the common output until the clutch is re-engaged. This is effected electrically by means of remotely operated solenoids which are operated from the cockpit. The real value of the remote re-engagement facility is to allow complete checking of separate control channels without the need for complex test equipment.

Each actuator has electrical feedback. In addition a low-gain mechanical feedback centers the actuator in the event of the loss of electrical power. The actuator centers automatically when either hydraulic supply is on, independent of electrical power. This action is equivalent to mechanical spring centering which is the conventional but heavier method. The mechanical feedback applies enough force to the flapper of the servovalve to cause the actuator to return to the midposition. The gain of the mechanical feedback is such that the performance of the actuator is dominated by the electrical feedback loop.

The configuration has the following advantages:

(1) Failure isolation is maintained

(2) Mechanization flexible with respect to redundancy

(3) Servovalve spool transducers not required

(4) Channel transfer is very fast because electrohydraulic solenoids are not used

The configuration has the following disadvantages:

(1) Output deviation (although very small) required for failure detection

(2) Force-voting mechanism will tend to get large and heavy for high power actuators

(3) Depends on voting mechanism reliability for transfer

CONCEPT 20-005 - (continued)

Comments:

By employing the output member as a secondary actuator, disadvantages 1 and 2 would be eliminated.

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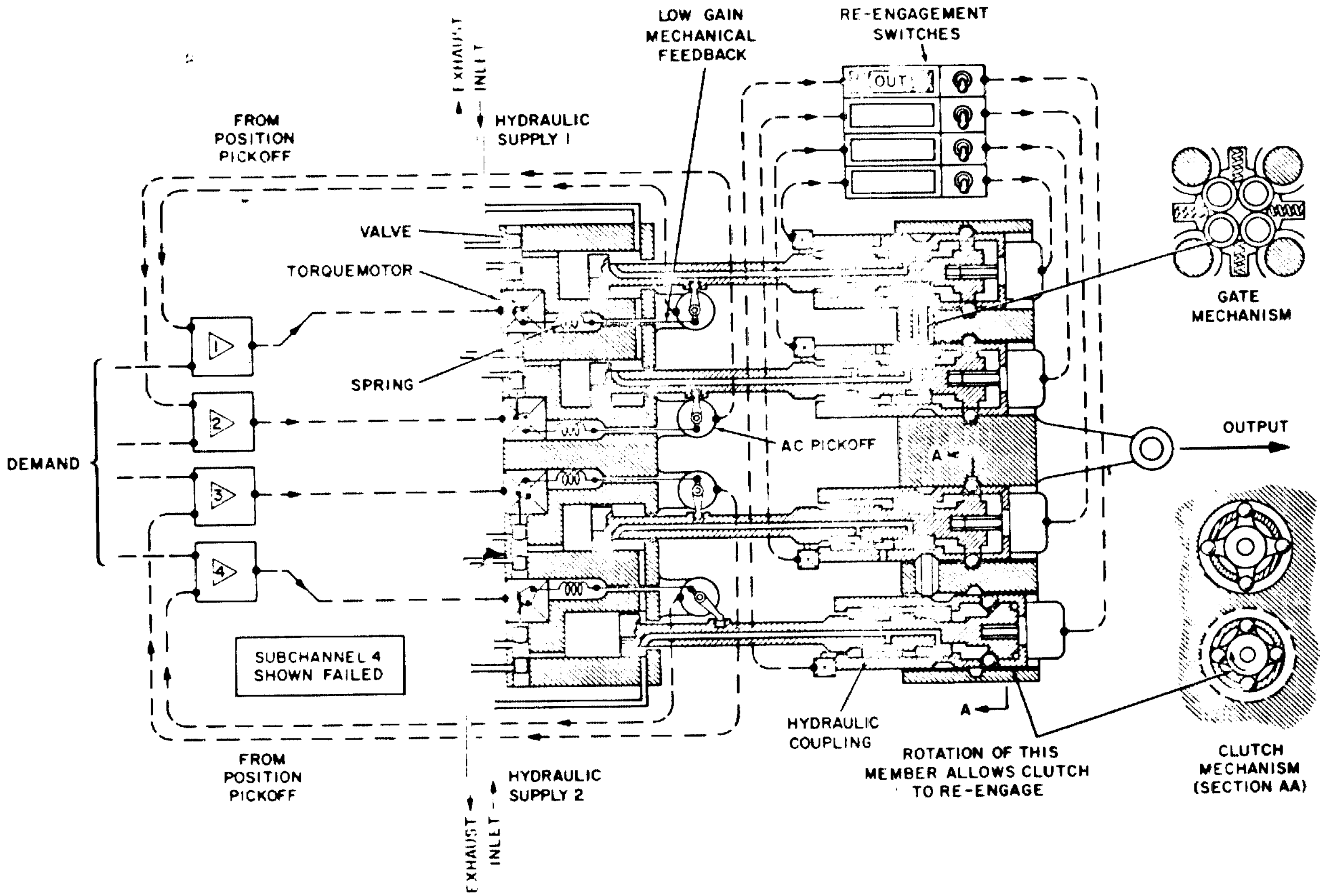

FIGURE A-34 FORCE SUMMED VOTED ACTUATOR

CONCEPT 21-018 FBW SYSTEM

The McDonnell Douglas FBW system was designed as a laboratory model demonstrator of a single axis redundant fly-by-wire flight control system.

The objective of the McDonnell Douglas FBW program was to design, fabricate, and evaluate a fly-by-wire flight control system compatible with the flight control requirements of aircraft of advanced design. An additional requirement was that the system offer a potential for significant improvement in reliability over currently proposed electrical flight control systems.

The triple redundant FBW flight control system is an electrohydraulic single fail operate, median select mechanization which controls the position of an actuator in response to flight control commands originating at a pilot's control stick. The principal parts of the system are: an electrohydraulic servoactuator which can accept three electrical signal inputs, three channels of electronic demodulators and servo amplifiers and four sets of LVDT triple tandem position transducers.

CONCEPT 22-149 FOUR CHANNEL FORCE SUMMING SYNCHRONIZED ACTUATOR

The secondary actuator consists of four distinct channels connected to a commom mechanical output as shown in Figure A-35. Each channel uses an electrical position transducer to create a position servo. The input to each channel is a position command in electrical form.

A Δ P transducer is used in each channel to sense the channel load pressure. The load pressure of each channel is compared to the average load pressure of all active channels and the differences are used as feedbacks to drive the individual channels towards the average load pressure. This provides synchronization of channel load forces.

The authority of the load synchronization in each channel is limited to a definite value. When this value is reached, the synchronization loop is unable to keep any further system or input unbalances from raising the channel load pressure and as a result it will increase to a second preset level which will cause a channel failure signal. This failure signal is used to disengage the channel.

A module is a self-contained, miniature servo system as shown in Figure A-36. A servo valve supplies flow to a small actuator which through a closely fitted pin on the drive arm converts linear motion to rotary motion.

The loop is closed through a LVDT from the same drive arm.

In normal mode of operation, hydraulic pressure is applied to the cylindrical drive piston which pushes the drive arm against a cam. In this manner the drive arm is coupled to the output shaft without lost motion. The application of hydraulic pressure to the drive piston is monitored by a pressure switch, which can be used to energize a pilot light.

The output shaft is supported by two bearings and is sealed by two rotary teflon seals. The two ends of the shaft are different, one is equipped with two prongs which serve as detent springs. The other end is equipped with a square hole. The prongs of the next module, in line, fit into the square hole. In this manner a latchless joint between modules is obtained. The spring detent is designed to transmit normal operating torques only, however, should excessive load be encountered the springs are deflected and a solid portion between the prongs makes contact in the square hole and the modules are capable of high torques for chip shearing or similar problems.

When a failure, through the Δ P sensing is detected, the solenoid is de-energized and the force is removed from the drive pistion. The drive arm is then free to disengage the cam and free itself from the output shaft. The remaining modules continue to drive through without additional loads.

CONCEPT 22-149 - (Continued)

When the drive arm disengages the cam it does not disengage from the actuator nor the LVDT. Should failure correct itself or should the pilot decide to re-engage, energization of the solenoid is all that is necessary.

An additional feature of the cam is: should the solenoid fail the drive piston can be forced to disengage the cam without freeing the remaining modules.

Since modules are only coupled through the output shafts, complete structural and hydraulic separation exist.

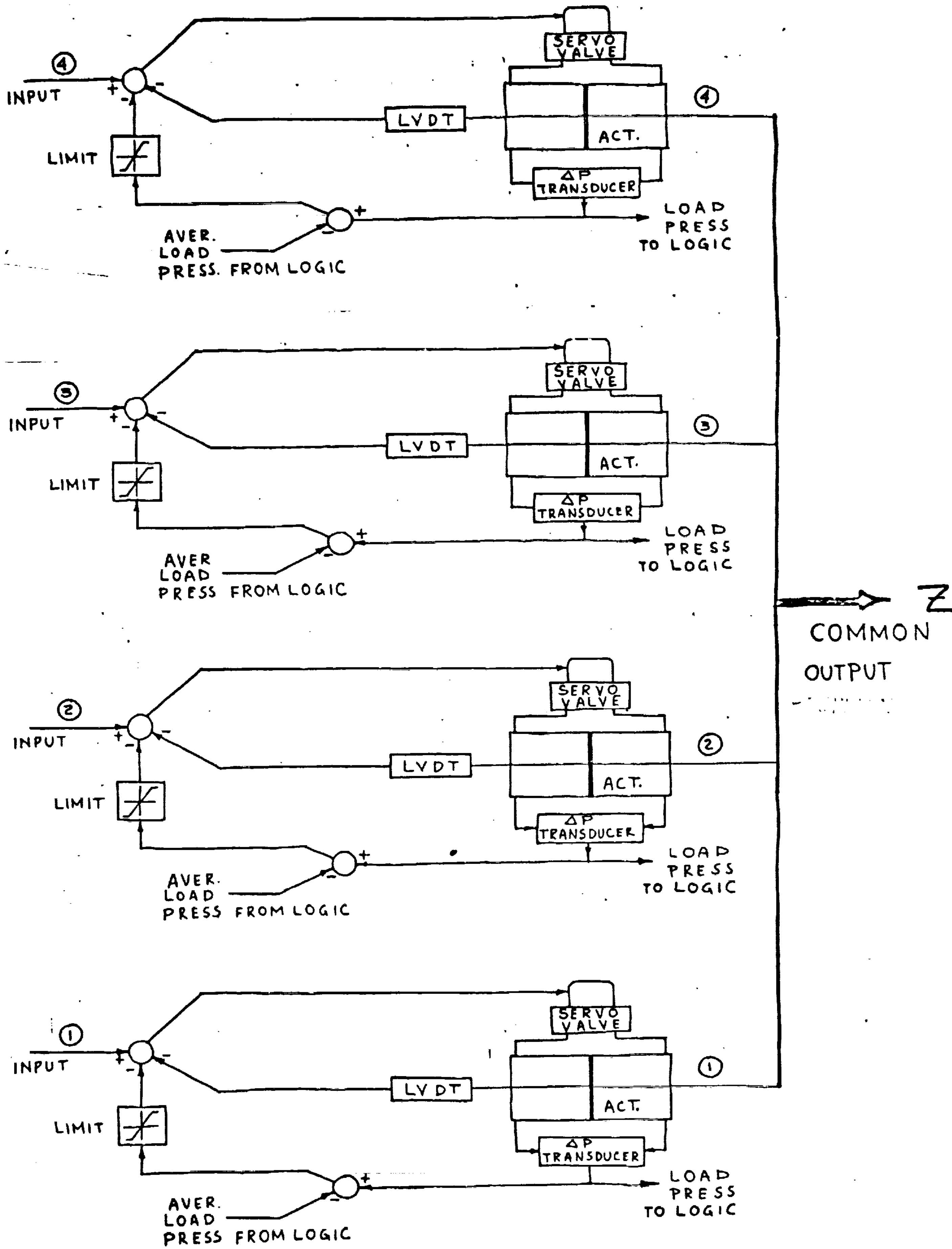

FIGURE A-35 FOUR CHANNEL FORCE SUMMING SYNCHRONIZED ACTUATOR ARRANGEMENT

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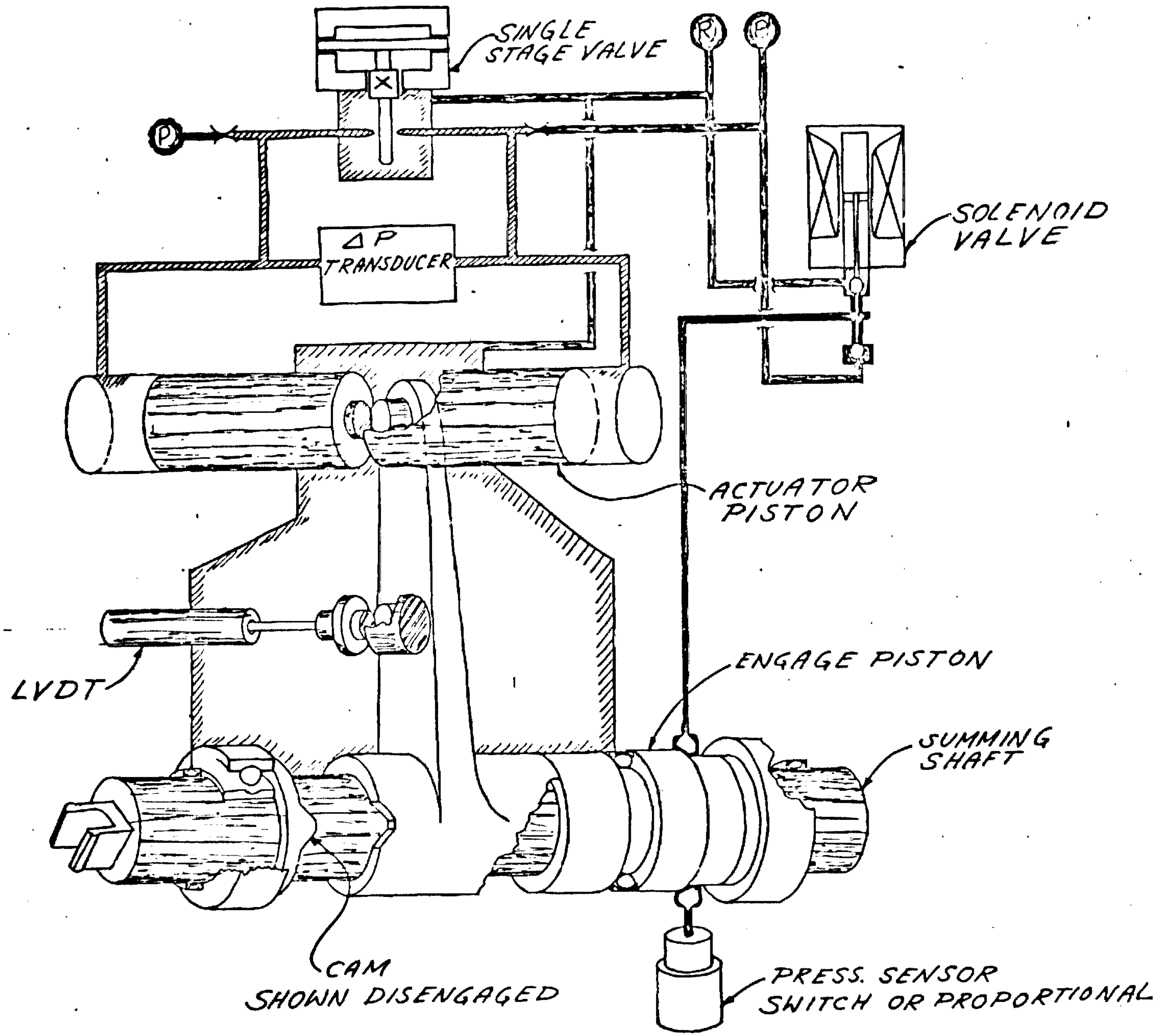

FIGURE A-36 FLY-BY-WIRE MODULE

This servoactuator is an electro-hydraulic, three-channel, force-summing configuration developed by Hydraulic Research and Manufacturing Company (HRM). The two-fail/operate, fail/passive capability is an implementation of redundant hydraulic control employing an intrasystem monitoring design.

A modular design approach is used to provide the required redundancy. This actuator consists of three independent systems or modules with complete hydraulic isolation controlling a triple tandem piston. All systems control the actuator at any one time. When a malfunction occurs in any system, that system is blocked by a shutoff/bypass valve, and the force output of the actuator is decreased proportionally. The actuator piston for that system goes into a by-pass mode.

Differential pressure transducers are used to provide feedback information for an electrical pressure equalization circuit. The pressure transducer located at each chamber of the actuator generates zero voltage at zero differential pressure and 20 millivolts at 3,000 psi differential pressure. The differential pressure feedback reduces the pressure gain (per system) from 6,000 to 9,000 psi per milliampere to approximately 750 psi per milliampere. This gain reduction reduces the deadband to eliminate force fighting.

After hydraulic pressure is applied (Figure A-37), the three solenoid valves are pulsed to engage the actuator. Once pulsed, the solenoid valve is held on the seat with system hydraulic pressure. This pressure drives three shutoff valves against their springs and activates the three systems. The active servovalve in each system controls the actuator.

The servovalves consist of an electrical torque motor and a hydraulic output stage. The output stage of this two-stage valve is closed center, which means that the spool is designed to block fluid flow when at the null position. Current flowing in the torque motor coils induces a torque in the armature, which pivots the flapper slightly toward either nozzle. This motion unbalances the hydraulic amplifier circuit, causing a pressure difference to be generated between the two end chambers of the second stage spool. This pressure difference creates motion in the second stage spool which varies the flow metering area in the sleeve, thus, changing the output flow.

Flow proportional to input current is achieved by the use of rectangular metering slots, and by restraining the spool with a feedback spring referenced directly to the torque motor armature. This mechanical feedback feature produces a torque on the armature proportional to the spool displacement. The torque transmitted to the armature by the feedback spring opposes the torque induced by the input current. Equilibrium results in spool displacement and flow proportional to input current.

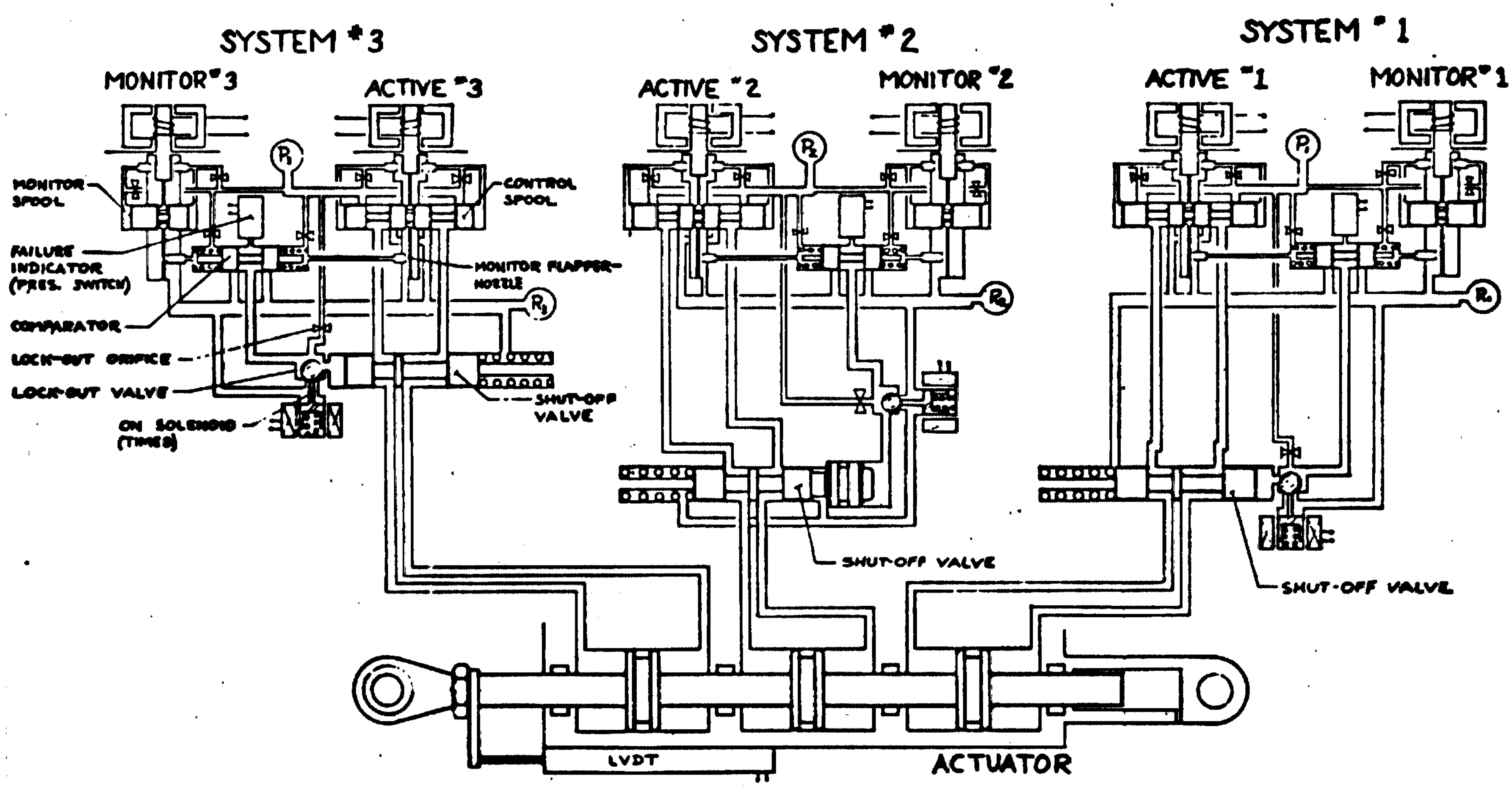

FIGURE A-37 HYDRAULIC SCHEMATIC - ARM C

The valve is modified with the addition of a second stage monitor flapper nozzle (Figure A-37). The function of the monitor flapper nozzle is to develop pressures proportional to the position of the second stage spools of the active and monitor valves. These two pressures are fed to opposite ends of the comparator spool. If no malfunction occurs these two pressures will vary but will remain equal in magnitude and the comparator spool will remain centered.

If a malfunction occurs the outputs of the active and monitor valves will differ. This will cause a pressure difference on the comparator spool creating motion of the spool. When the pressure difference exceeds a predetermined threshold, motion of the comparator spool will dump the supply pressure, holding the shutoff valve, to return. The shutoff valve of the failed system will be forced by the spring pressure into a bypass position. The bypass position blocks the output of the active servovalve of the failed system. The actuator will continue to operate with the remaining controlling systems.

The failure threshold of the comparator can be easily varied by spring rate on, and overlap of, the comparator spool. Once the optimum threshold is determined it will remain fixed.

If a malfunction occurs in a second system it will be placed into a bypass mode. The remaining system will continue to control the actuator. The sequence of system failure is no problem. All systems are operational and only a failed system is switched out.

A third failure will cause the actuator to fail in a bypass mode on all three systems. System failure is detected by a pressure switch on the comparator valve.

Pressure loss, exceeding a predetermined threshold in any system, will cause the ball in the solenoid valve to unseat, thus, switching out that system.

After a malfunction, a failed system will not come back on line until the solenoid valve is pulsed. If the malfunction has been corrected, pressure will hold the solenoid valve ball on its seat, the input to the comparator spool from the active and monitor valves will be identical, the shutoff valve will be pressurized, and the pressure switch will cycle, thus, returning the system to normal operation. If the malfunction is still present, the system will immediately switch out as before.

Attached to the actuator output are four position feedback linear variable differential transducers (LVDT's) (Figure A-38). One LVDT is dedicated to each of the three systems for servo stabilization and all four LVDT signals are sent to failure detection logic. This logic uses a cross-channel failure detection method. Each LVDT signal is compared with the signals from all other working LVDT's. A fail decision is made if the signal of that LVDT differs appreciably from that of the other LVDT's. The failure threshold is an error voltage equal to that generated by displacing the actuator five percent of full travel. The detection of a failure energizes a latching relay which provides a positive d.c. vias voltage to the monitor servo-amplifier. This causes the hydraulic logic to disengage the failed channel.

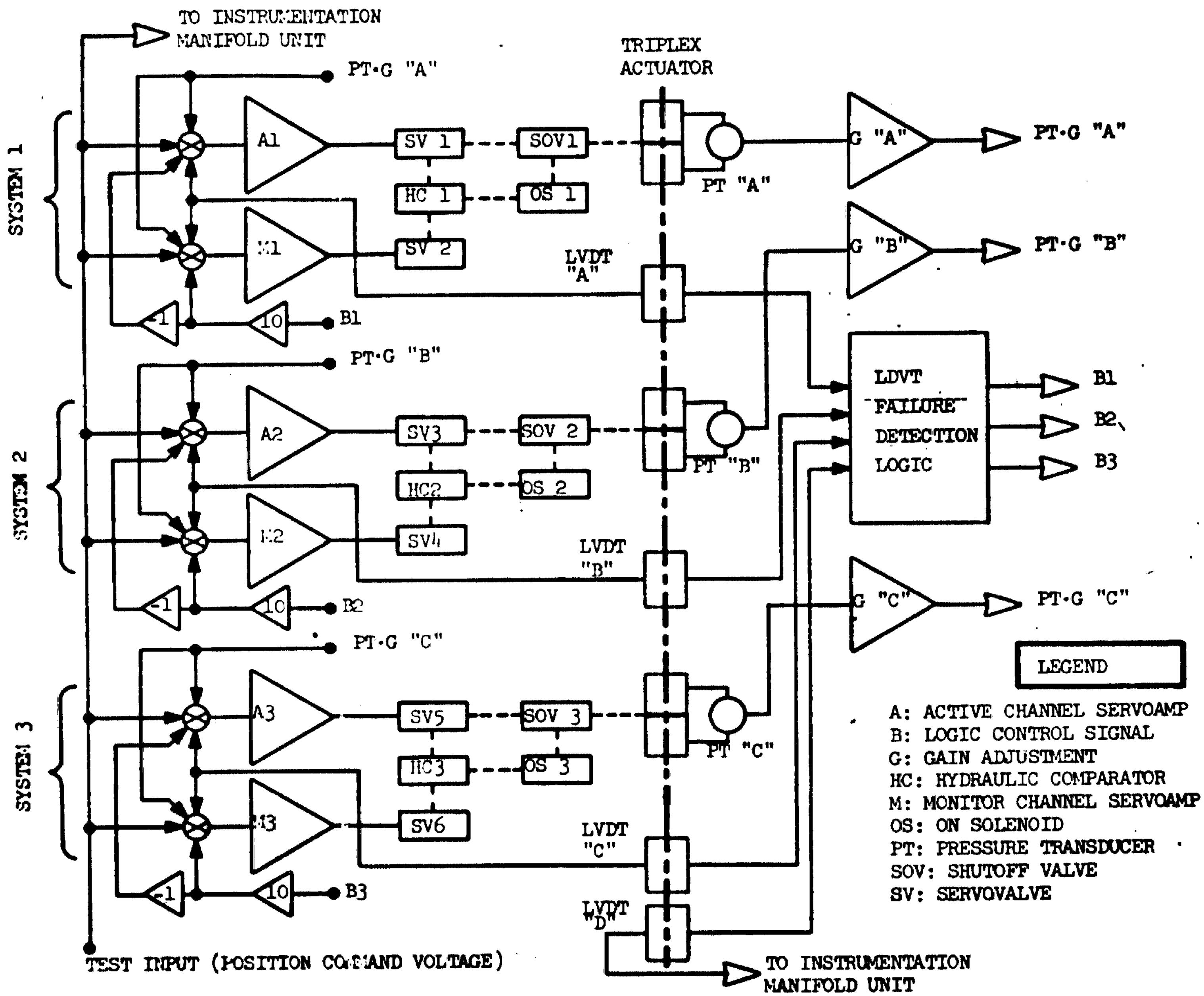

FIGURE A-38 ELECTRICAL SCHEMATIC - ARM C

CONCEPT 24-009 F111 SERVOACTUATOR

The dual tandem actuator with hydromechanical model, which is used on the F-111, remains undegraded by the first fault (Figure A-39). It consists of two actuator sections in tandem, a differential sensor spool, and a hydromechanical model. The spool is used to balance the load between the two actuator sections and also, together with the model, for automatic fault removal. When the displacement of the output shaft differs from that of the model piston by a predetermined amount, the direction of the difference is compared with the spool position to determine the faulty section, which is then deactivated. The other section continues to operate with the same gain and basically the same dynamic performance as before.

The performance capabilities of redundant actuator configurations can be expressed in terms of the transient response to a single fault. Assuming an actuator shaft travel of ± 0.75 in., a single hardover fault in a servo valve, for example, produces an output shaft transient of 0.18 in., which lasts less than 0.16 sec, in the dual side-by-side actuator and a 0.20 sec peak transient of 0.26 in. in the dual tandem actuator with hydromechanical model.

The penalties in fabricability and maintainability that must be paid for increased performance mainly are the obvious ones. The difficulty of fabrication generally depends on the tolerances that must be held on components and on the complexity of the subassemblies. A servo valve that requires mechanical feedback from one or two points, for example, is more likely to prove troublesome than one that requires no such feedback. In the latter case though, whatever other arrangement is chosen for the feedback function may lead to a net increase in complexity. Similarly, it is not necessarily more difficult to fabricate a complex valve in a single unit than to put together an assembly of several less complex components.

In assessing the test requirements, it has to be kept in mind that all actuator functions must be exercised and checked at the beginning of each flight or maintenance period. This includes the fault removal functions, for a part used only in fault removal may well have failed in a normal position. The degree of maintenance testing thus tends to correspond to the degree of complexity.

Similarly, repairability in general is inversely proportional to complexity. Any part that can be replaced without alignment or trimming is unlikely to cause problems. If changing a servo valve means merely pulling out and driving in four screws, the job is simple. If electric feedback is used, however, the null position of the pickoff must be adjusted, and matters have become more complicated. Also, in the case of the more complex valves, larger assemblies must be replaceable as units, and special equipment is needed for repairing these assemblies. As all their parts must be repaired or replaced in the event of a fault, the total number of piece parts in a given actuator can be used as a good measure of the required maintenance repair effort.

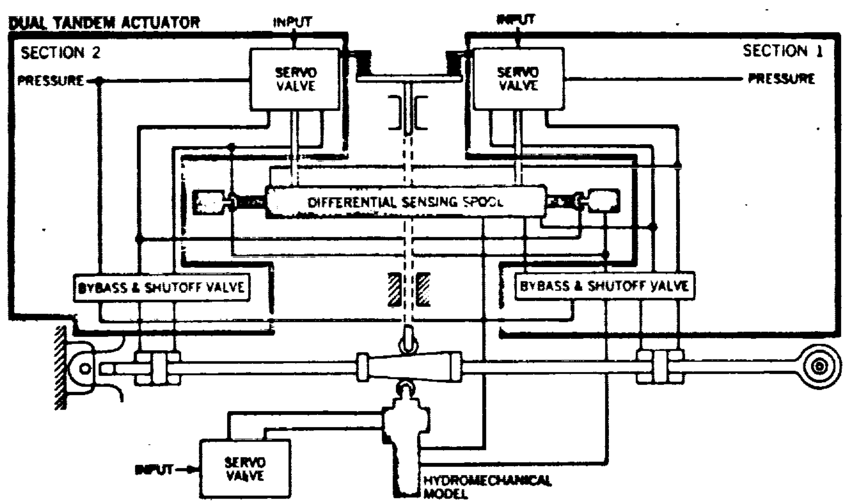

FIGURE A-39 DUAL TANDEM ACTUATOR

CONCEPT 25-092 FORCE SUMMING WITH FIVE ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP 680J

In this system five electromechanical actuators drive the main valve simultaneously. (Figure A-40). The output of each actuator is transmitted to the main servo valve through a funk spring. Funk spring breakout force is equal for all actuators, but is made higher in one direction than the other to permit positive control under all conditions. Should any two of the actuators fail in any combinations or sequence of jams and opens, the remaining three combined have the capability of driving the main valve by breaking out the funk springs of the failed units. The difference in force required to break out a funk spring in one direction as opposed to the other assures that at least one of the remaining units is operating within its detent range. A transient effect is produced in reaching the new position where at least one spring will be in its detent range after a failure; however, the system retains it stiffness no matter what the failure.

In the event of a failure in one channel, the remaining channels control the output and the actuator in the failed channel causes its funk spring to collapse and the cut-out or failure switch to be activated. The switch in turn removes power from the failed channel and allows the failed motor to be back-driven by the remaining channels through the high-efficiency ball screw gearing. In the event of a jam in a motor, the remaining channels breakout the funk spring in the failed channel and simply over power the failed unit.

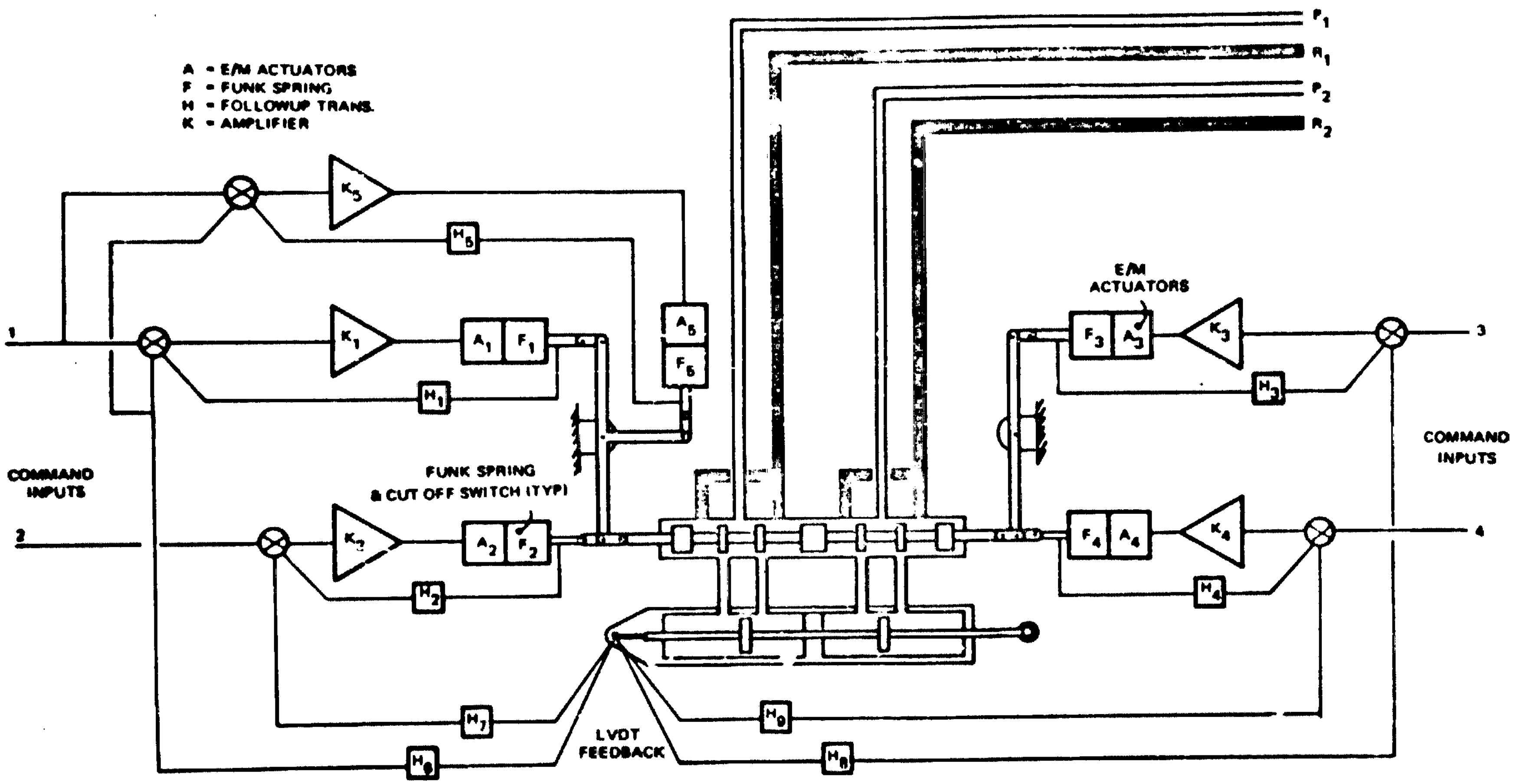

FIGURE A-40 FORCE SUMMING WITH FIVE ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP

CONCEPT 26 -092 FORCE SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP 680J

This system operates identically to Concept 57-092 with the exception that only one jammed E/M actuator can be tolerated, Figure A-41. This is considered a realistic limitation and eliminates the requirement that all four signal channels must be good. Two hard over signal failures are still allowable as each channel can be back driven after electrical disconnection. The presence of only four E/M units (instead of five) allows for a greater funk strut breakout force differential between the two directions of operation. This results in a greater force margin for control of the main servo valve.

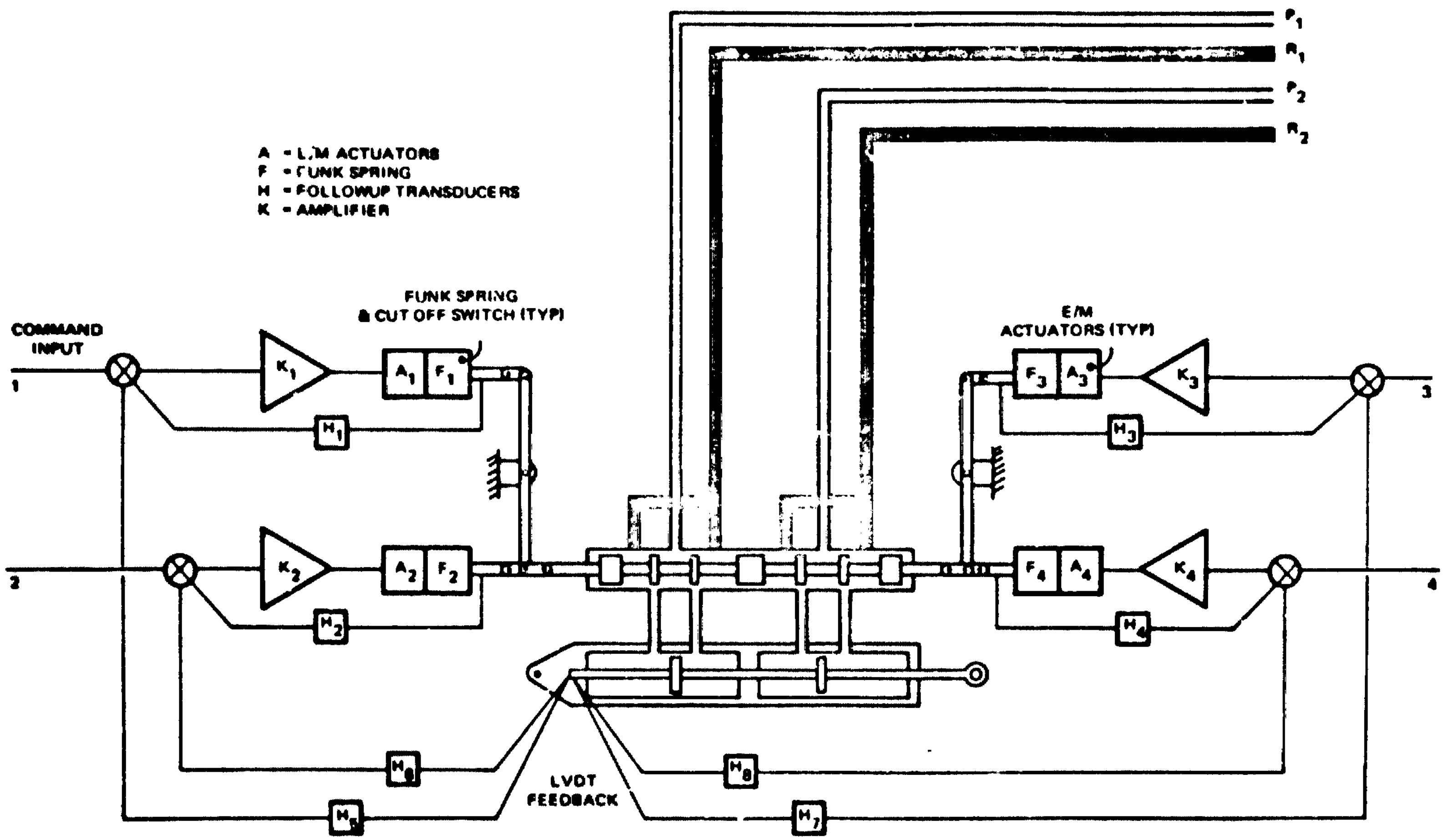

FIGURE A-41 FORCE SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP.

CONCEPT 27-092 FORCE SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP 680J

This system is identical to Concept 27-092 with the exception that the feedback of the main actuator position is mechanical rather than electrical, Figure A-42. This method of position feedback places the four electromechanical actuators outside the (main actuator) feedback loop, reduces the electrical feedback complexity, but introduces more severe transient and trim effects. In addition, long stroke E/M actuators become necessary.

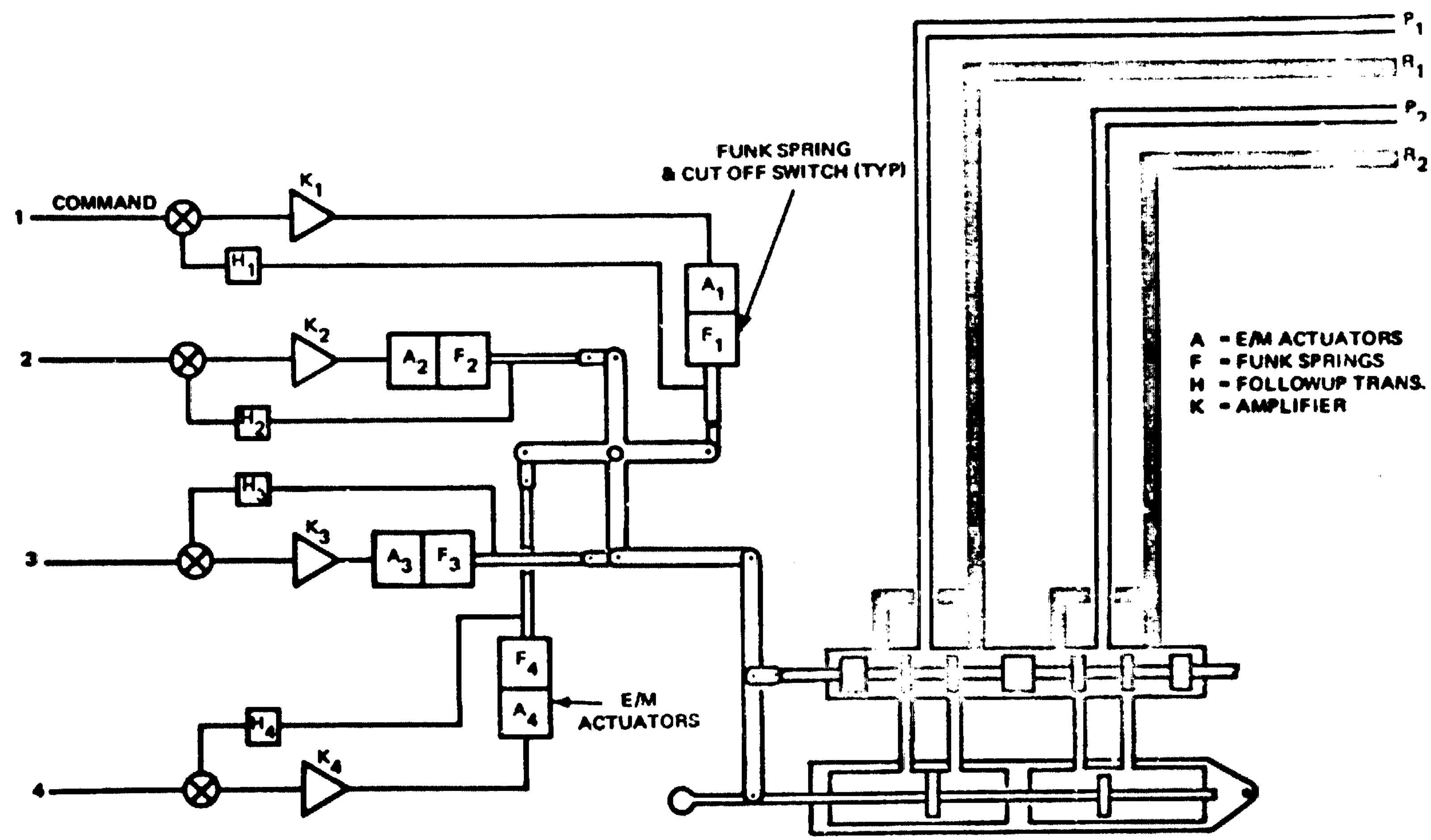

FIGURE A-42 FORCE SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP.

CONCEPT 28-123 MAJORITY VOTING SERVOACTUATOR

The basic design approach with the majority voting servoactuator is to triplicate all critical components in the torque motor, hydraulic amplifier and feedback portions of the unit. Tight inner-loop feedback is then used to reduce the change in performance when a failure is present. This inner-loop is formed by feedback of valve spool position, so the actuator uses, in effect, a majority voting servovalve (see Figure A-43. Three conventional torque motors are used each having a separate armature and flapper assembly. Each flapper controls flow from two opposing nozzles which receive fluid through fixed inlet orifices. The differential flow from the three hydraulic amplifiers supplies the end areas of a free-floating valve spool. The maximum differential flow from any one hydraulic amplifier is limited by the physical bottoming of its flapper against one nozzle tip. Whenever a net differential flow exists, the valve spool moves. As it moves, a corresponding feedback torque is created on each armature/flapper by separate feedback springs.

This combination of three or more separate controls, each with a limit on its maximum output, where the three outputs are summed, and with feedback about each control channel, forms a basic majority voting configuration. If a hardover condition develops in one channel, either through failure of a component or from an erroneous input, the output will start to follow. But as the output changes compensating feedback is developed at the remaining two good channels. The magnitude of output offset with a single hardover failure is related to the feedback gain and to the separation of the limits. Typically this offset will be less than 5% with a hardover failure in the majority voting servoactuator.

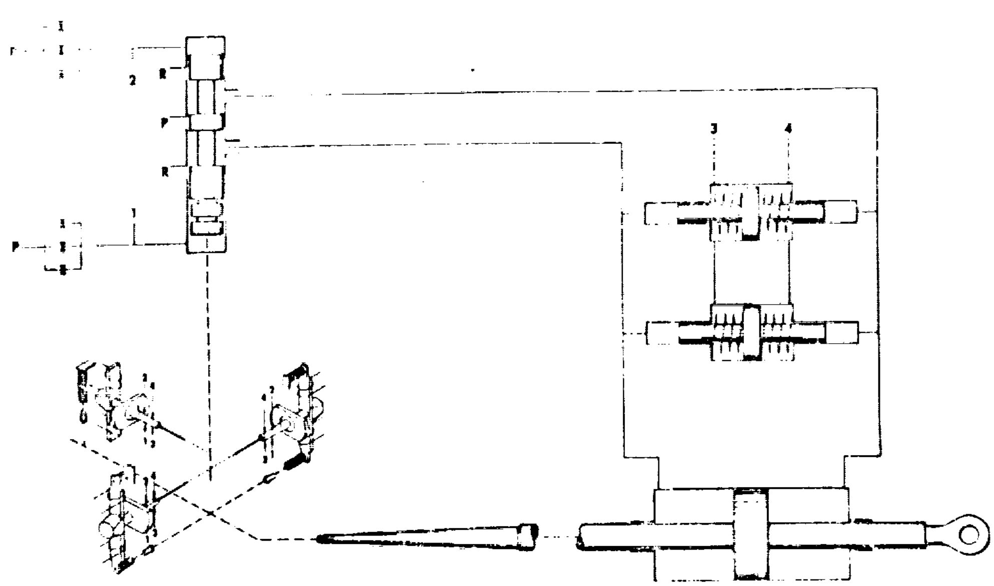

FIGURE A-43 PICTORIAL SCHEMATIC MAJORITY VOTING SERVOACTUATOR

CONCEPT 29-009 A7A SERVOACTUATOR

The dual fail-off actuator (Figure A-44) consists of two active actuator channels side by side. Each channel actually is an independent actuator with its own piston, cylinder, and electrically driven two-stage hydraulic servo valves, and each receives an electric signal from its own servo amplifier. The two pistons are connected to the output shaft so that there is no relative motion between them, while the two cylinders are connected to each other through a center-pivoted differential link attached to the main frame of the actuator.

The load is shared by the two channels, so that the differential link does not move when the flow is the same in both channels but does move even in the presence of small flow variations due to normal parts tolerances. The output shaft motion is the average of the motions of the two channels.

The outputs of both channels are monitored by an electronic comparator, and the actuator is center-locked if the channels differ by more than a predetermined amount. To limit the maximum difference and so the output transients produced by a fault, the differential link is allowed only limited mechanical travel.

The same basic effect could be achieved by using two nonredundant actuators and summing their outputs in the airframe control run. The dual fail-off actuator, however, has a smaller transient when a fault occurs, weighs less, and takes up less space.

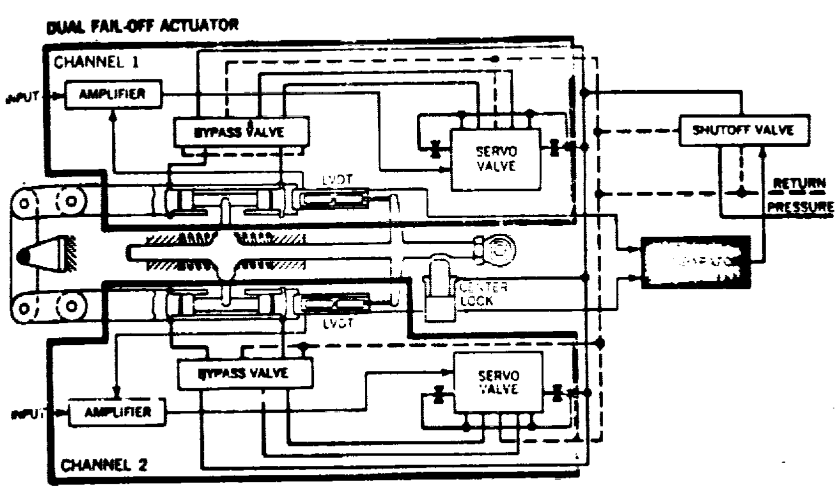

FIGURE A-44

CONCEPT 30-009 SERVOACTUATOR

The dual side-by-side actuator with electronic model is very much like the dual fail-off design, except that a second shutoff valve, differential link locks, and an electronic model have been added and the simple electronic comparator has been replaced by a majority-logic unit (Figure A-45). The model is a solid-state demodulator and RC network that supplies a dc analog voltage to the comparator. The input voltage to the model is the same as that to the servo amplifiers, and the circuit constants of the model are sized to provide the same gain and time constants as do the servo amplifiers, hydraulic actuators, LVDTs (linear variable differential transformers, used as follow-up sensors), and demodulator chains in the active actuator channels. The output of the model therefore is statically and dynamically similar to the rectified output of the LVDTs.

The logic networks compare the output of the model with that of the position pickoffs of the channels. When a fault occurs, they detect the affected channel and deactivate it; when a second fault occurs, they center-lock the actuator.

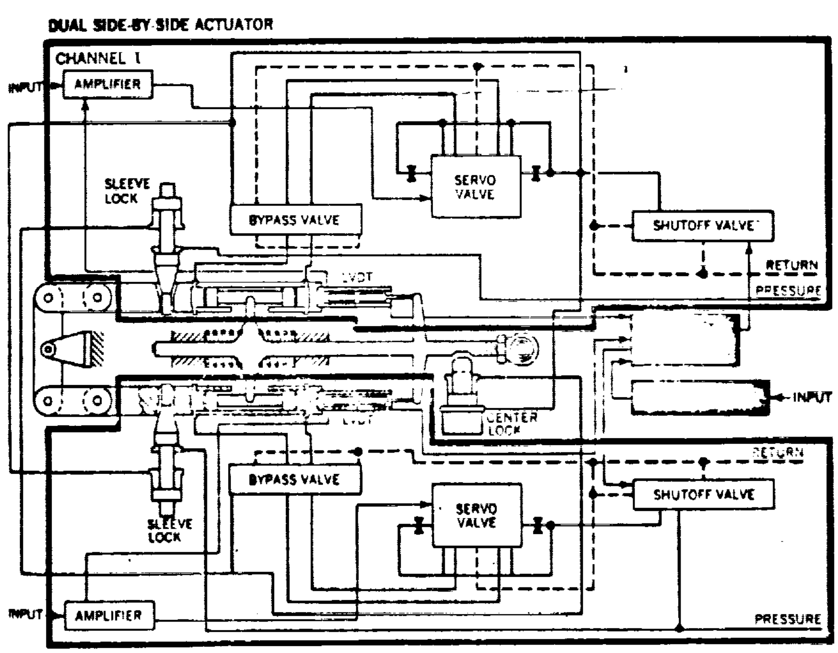

FIGURE A-45

CONCEPT 31-142 680J POSITION SUMMING SECONDARY ACTUATOR

An example of this type of mechanization is shown by the schematic in Figure A-46. Here the actuator is dual redundant and it is combined with an electronic model to obtain a single failure correcting actuator with centerlock upon second failure. The actuator consists of two active elements in a side by side arrangement. The pistons of both channels are connected rigidly to the output so that no relative motion occurs between them. The cylinders are connected to a center pivoted differential link so that the output motion is the sum or average of the two actuator motions. If the output of element 1 is A and the output of element 2 is B, the actuator output is (A + B)/2. During normal operation when both elements are positioned a distance X the output of the actuator is X. The differential link motion is limited by stops to a level slightly above the failure detection threshold. If a failure occurred in element 2, (B), the output of the actuator is (O + B)/2 and the maximum output is limited by the allowable differential link motion. Assuming that the position outputs of the elements are compared in a cross element comparator-monitoring scheme, the failure in element 2 can be detected and the element can be shut down and bypassed through the solenoid operated shutoff valve. From the schematic it can be seen that when element 2 is bypassed and the cylinder and differential link is locked, then element number 1 is in command of the output and no degradation in output position or velocity occurs since the piston of element 2 is free to move with the output.

A quadruplex position summing actuator is presented in schematic form in Figure A-47. The mechanization illustrated consists of a pair of the dual redundant actuators. The actuator is capable of operation after two failures with no degradation of output position or velocity performance, and the output can be centerlocked or braked near the last position after a third failure.

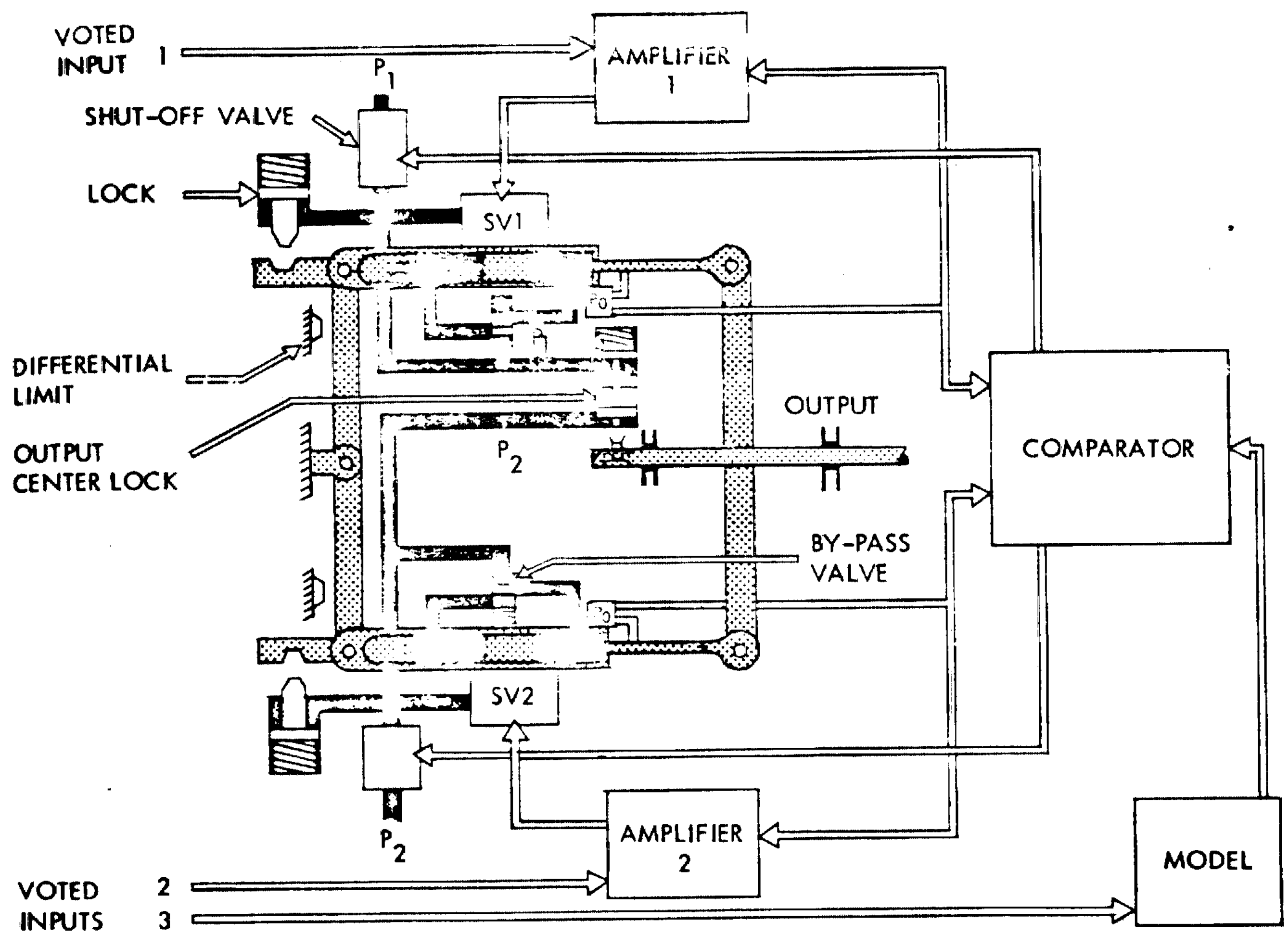

FIGURE A-46 POSITION SUMMING FAIL OPERATE/FAIL CENTER ACTUATOR

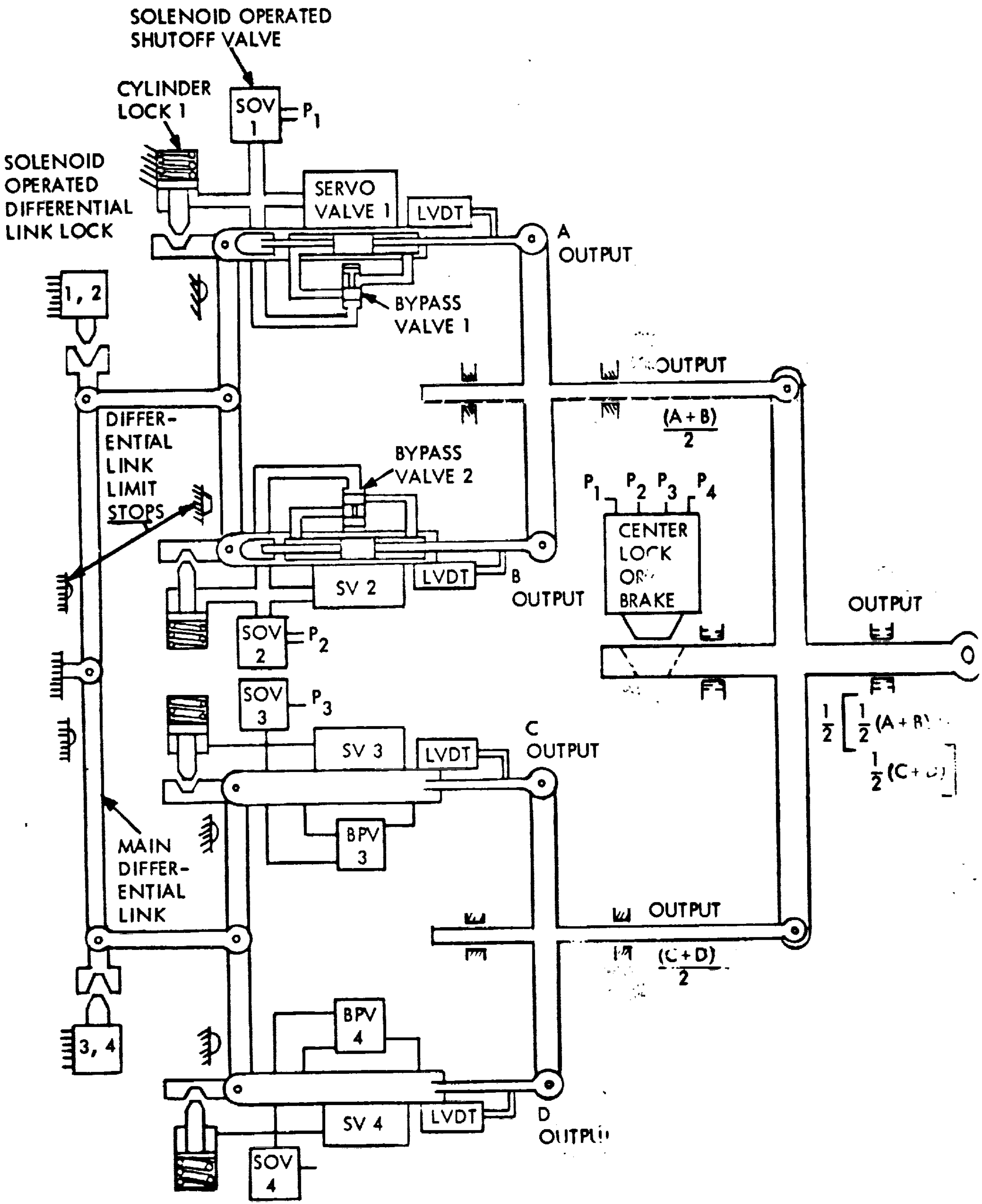

FIGURE A-47 QUADRUPLEX POSITION SUMMING ACTUATOR

CONCEPT 32-039 ELECTRO-MECHANICAL REDUNDANT ACTUATOR MECHANISM

The Electro-RAM (Electro-mechanical Redundant Actuator Mechanism) was conceived and developed by LTV Electrosystems. Electro-RAM will be flown in the Air Force ADP 680J SFCS program as a driver for the dual-tandem hydraulic power servo that positions the stabilator of the F-4 aircraft. The power servo is known as the Survivable Stabilator Actuator Package (SSAP) and develops its own hydraulic power locally from two electrically driven hydraulic pumps integrated into the actuator. The Electro-RAM is also integral with the SSAP package where it converts four identical (nominally) electrical signals into a single position command to the power servo. The four channels of Electro-RAM are continuously monitored resulting in two-fail-operate performance. Figure A-48A schematically depicts the role of Electro-RAM and the SSAP in controlling the stabilator of the F-4 aircraft. Figure A-48B is a drawing of the SSAP.

Electro-RAM uses four brushless A-C servomotors connected in pairs through mechanical differentials, and the output velocity is the sum of the motor speeds. This is active redundancy contrasted with standby redundancy where only one channel carries the load while the others stand by in the event of failure of the active channel. Figure A-48C is a mechanical schematic of the 4-channel Electro-RAM. The rotary motions of Electro-RAM are converted to linear motion by a highly efficient ballscrew. A second ballscrew is provided for output redundancy. The linear stroke is fed back to the four separate motor control servo amplifiers from a quadruple transducer. Velocity summing is a form of displacement summing with infinite stroke at the motors. The advantage of summing rotary motions is that the actuator delivers constant force and stroke no matter how many channels are operating. If a channel fails, power to that channel is automatically shut off and a motor brake is engaged. The other three motors continue to operate with no loss of force or stroke at 75 percent of original velocity. Subsequent failures result in similar 25 percent decreases in velocity but no loss of force or stroke.

The dual ballscrew design adds greater reliability. Should the primary ballscrew jam, the backup ballscrew automatically breaks out of detent and provides full output with no loss of motion or performance. No sensing or switching is required to transfer these motions.

Electro-RAM is monitored by comparison of tachometer signals from the motors. Logic is provided to detect a discrepant channel from mis-track between its tach and the good channel tachs. A failure discrete is generated which shuts off the discrepant channel and removes it from future comparisons.

Alternate embodiments of the basic velocity summing technique are available for other applications. For example, the design shown in Figure A-48D is appealing because it offers a greatly reduced parts count. Only three channels are provided with a rotary output. Ballscrews and two differentials are eliminated from the basic design. The motor motions are fed into the differentials through irreversible (essentially) worm drives thus obviating the brake requirement. If a channel fails, the only corrective action required is removal of power to that motor.

CONCEPT 32-039 (Continued)

Electro-RAM offers the FBW system designer a way of combining his redundant control system channels where he needs a single output. He can vote the channels at the actuator thus including all upstream elements in the vote. Because Electro-RAM uses active redundancy, it is efficient in size and weight, and failure transients are minimal. The characteristic of constant force and stroke regardless of the number of operating channels provides an extra measure of reliability. Also, the Electro-RAM makes an ideal Line Replaceable Unit (LRU) for ease of maintenance.

Electro-RAM is an example of the hardware being produced in current FBW development programs. A dual, three-channel flight control actuator system using the design of Figure A-48D has been shown to have a theoretical safety of flight reliability of .999,999,999,999,999,985.

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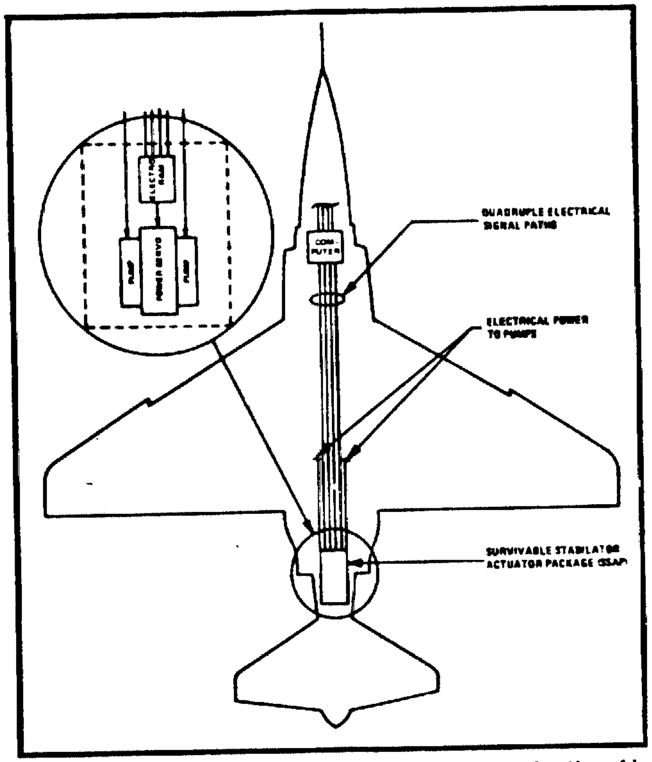
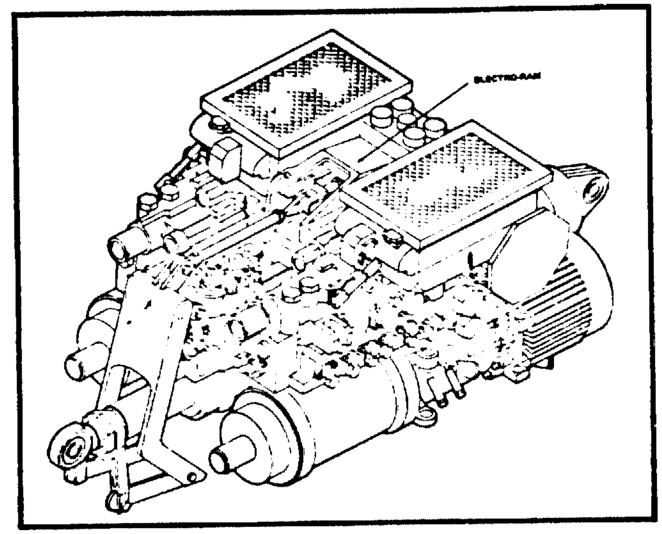
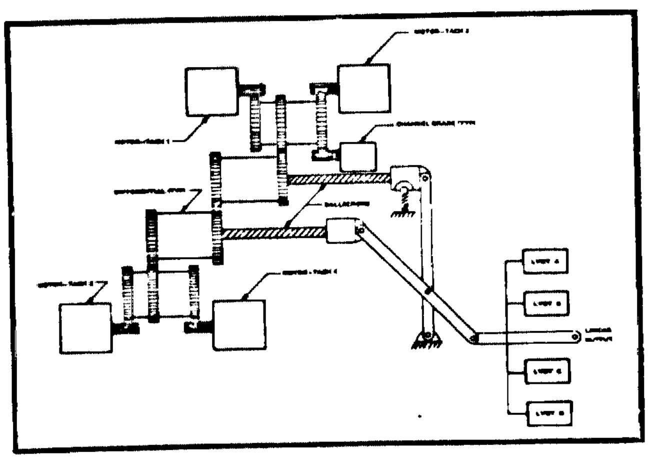


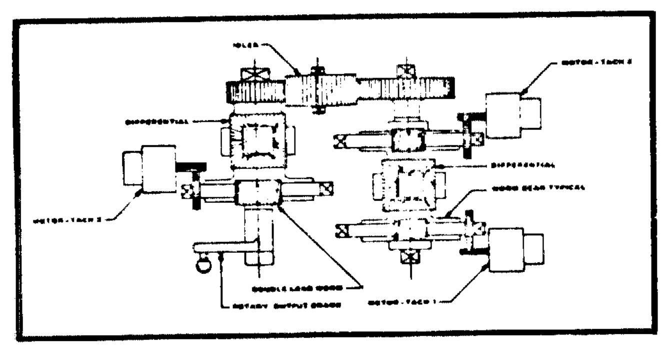
A. Electro-RAM and SSAP as Used in the Air Force "Survivable Flight Control System."



B. Survivable Stabilator Actuator Package (SSAP).



C. Schematic SFCS Electro-RAM 4 Channels.



D. Schematic Electro-RAM 3 Channels.

FIGURE A-48 THE ELECTRO-RAM

CONCEPT 33-092 DISPLACEMENT SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP 680J

In this system all four electromechanical actuators drive the main valve through a displacement summing linkage, Figure A-49. Two actuators are arranged to drive the valve slider and the other two drive the sleeve. The actuators are driven in opposite directions such that the result is a net relative displacement between sleeve and slider. A jam in any two actuators will not result in system failure unless they happen to be hard over jams in the same direction. A solenoid operated centering device is required to drive each failed actuator to center and lock. Electronic comparators are required to detect the position discrepancy and operate the required solenoid. Loss of two actuators results in loss of half the main servo valve output. Main actuator position, or LVDT output, is summed with the command signal of each E/M actuator.

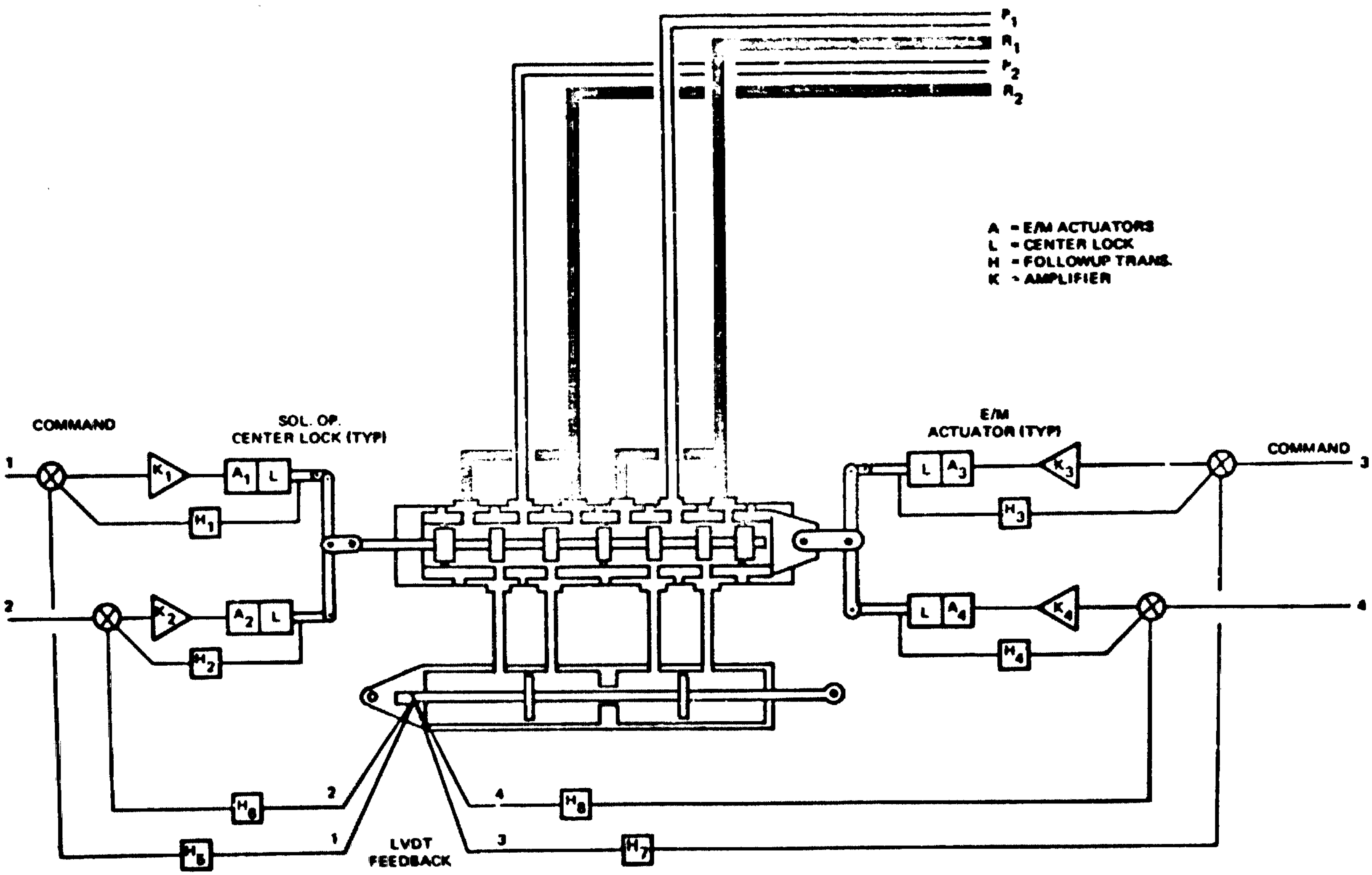


FIGURE A-49 DISPLACEMENT SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP

CONCEPT 34-092 DISPLACEMENT SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP 680J

The conversion technique used in this system is the same as that used for Concept 33-092. The difference between the two systems is that the signal converter in this system is outside rather than inside the loop, Figure A-50. This, of course, results in the requirement for mechanical feedback of the main actuator position. In addition, when displacement summing is employed outside of the loop, any failure results directly in the loss of position authority. This means that loss of one of four channels results in the loss of 1/4 of the output position capability.

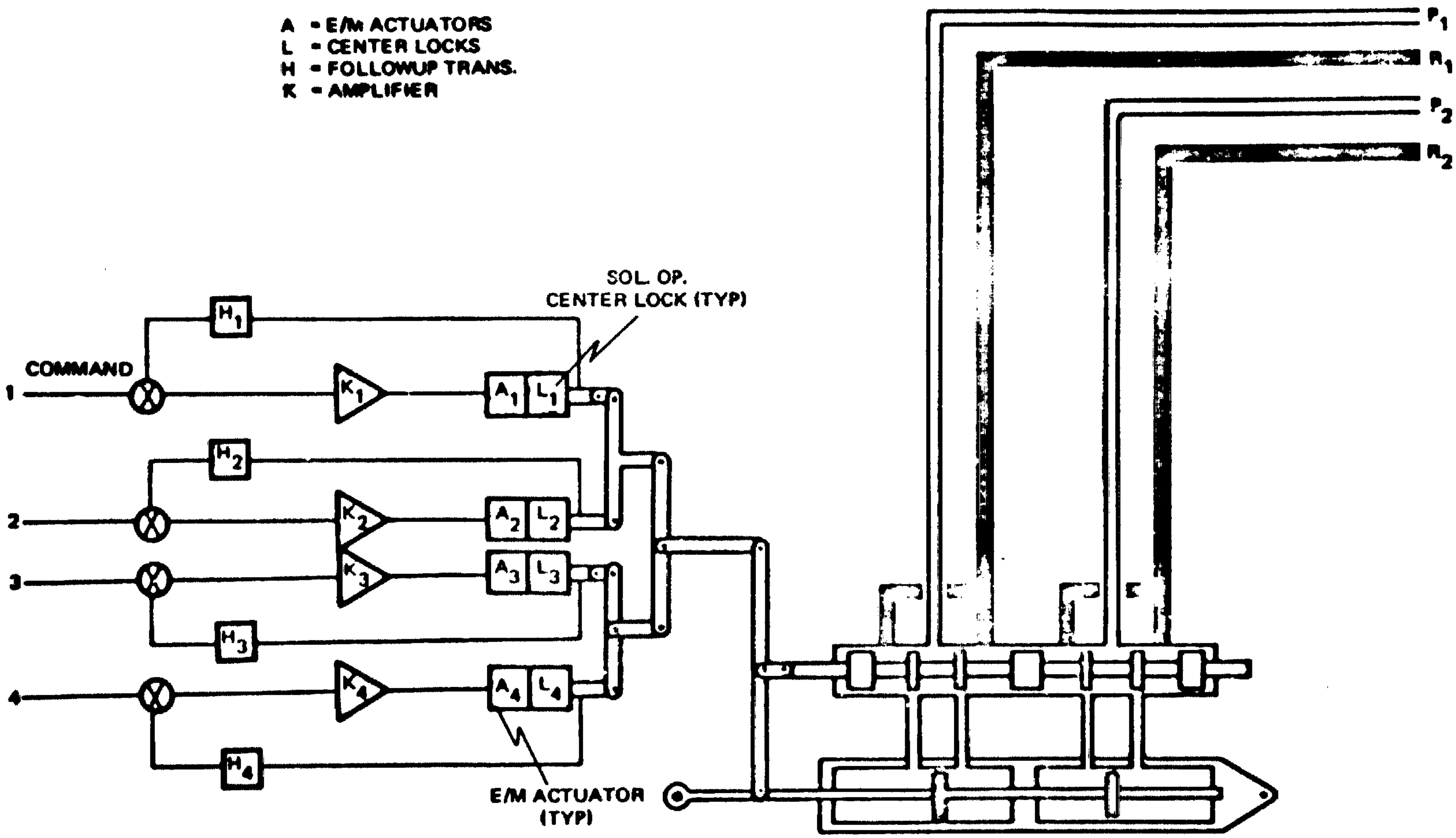


FIGURE A-50 DISPLACEMENT SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP

CONCEPT 35-092 DISPLACEMENT SUMMING WITH DIFFERENTIALS AND ELECTRICAL MOTORS INSIDE THE LOOP 680J

This system can be considered velocity summing or a displacement summing scheme which is essentially without the normal displacement limitations, Figure A-51. It is a rotational scheme which employs four electric motors whose velocities are summed through mechanical differentials and reversible gearing to drive the main valve. Under normal operating conditions, the valve sleeve is held stationary by a funk spring which restrains the output of differential C. Output sum of motors 1 and 2 are transmitted through differential A to one side of differential B. Similarly, the outputs of motors 3 and 4 are transmitted through differential D. However, because the output of differential C is restrained, the motion is transmitted across C to the other side of differential B to complete the summing action to the servo valve slider. Therefore, this system, unlike any linear displacement summing scheme, has an infinite stroke capability which has many advantages. One of these advantages is that full valve stroke can be achieved with as many as three of the four channels inoperative. Another important advantage is that failure monitoring and shut off can be achieved without electrical cross-coupling of the channels. This is possible because a constant velocity of the main servo valve is not germain to control of the aircraft, and can be washed out for normal operation and used for failure criterion. This is easily done in a rotational system.

Control of the motors is facilitated by an inner loop (4-channel LVDT measuring valve error) and an outer loop (4-channel LVDT on the main actuator piston). The inner loop is used to improve the system stability and to provide anticipatory failure information which reduces transient effects on the main actuators. A tach feedback closes the loop around each motor through a high gain lag circuit to keep the motors from circulating against each other due to missynchronization between the channels.

The main servo valve here is shown with redundant inputs. The normal action is for all displacements to be injected to the valve slider. (This is facilitated by the funk detent device on the sleeve). If a jam occurs in differential B, enough force is generated to overcome the funk detent and drive the sleeve. Jams in differentials A, C, or D result in normal slider motion, but with reduced velocity. Therefore, a jam in any single element does not result in failure of the system, and the system is capable of withstanding many jams with no more than a reduction in slider velocity. Open links become critical in displacement summing systems. The output of differential B can be mechanically limited such that it can bottom out and permit force to be applied at the sleeve input when the slider input is open. The slider is spring centered to keep it from floating along with the sleeve under this condition.

Should any motor fail, the associated brake is energized to ground the motor output at the failed position. The tach feedback output which is modified by the time lag circuit is fed to an error detection circuit. The error circuit is designed to shut off electrical power to the motor if the tach output reaches a critical level for a specified length of time. This could occur with such things as a broken feedback element, malfunctioning amplifier, or gross mis-synchronization. To protect against a failure of the techometer, the motor current is fed through a similar high gain lag circuit to the same error detector shut off. The outputs of the tach generators are available for comparison it it is considered necessary to detect an unresponsive motor.

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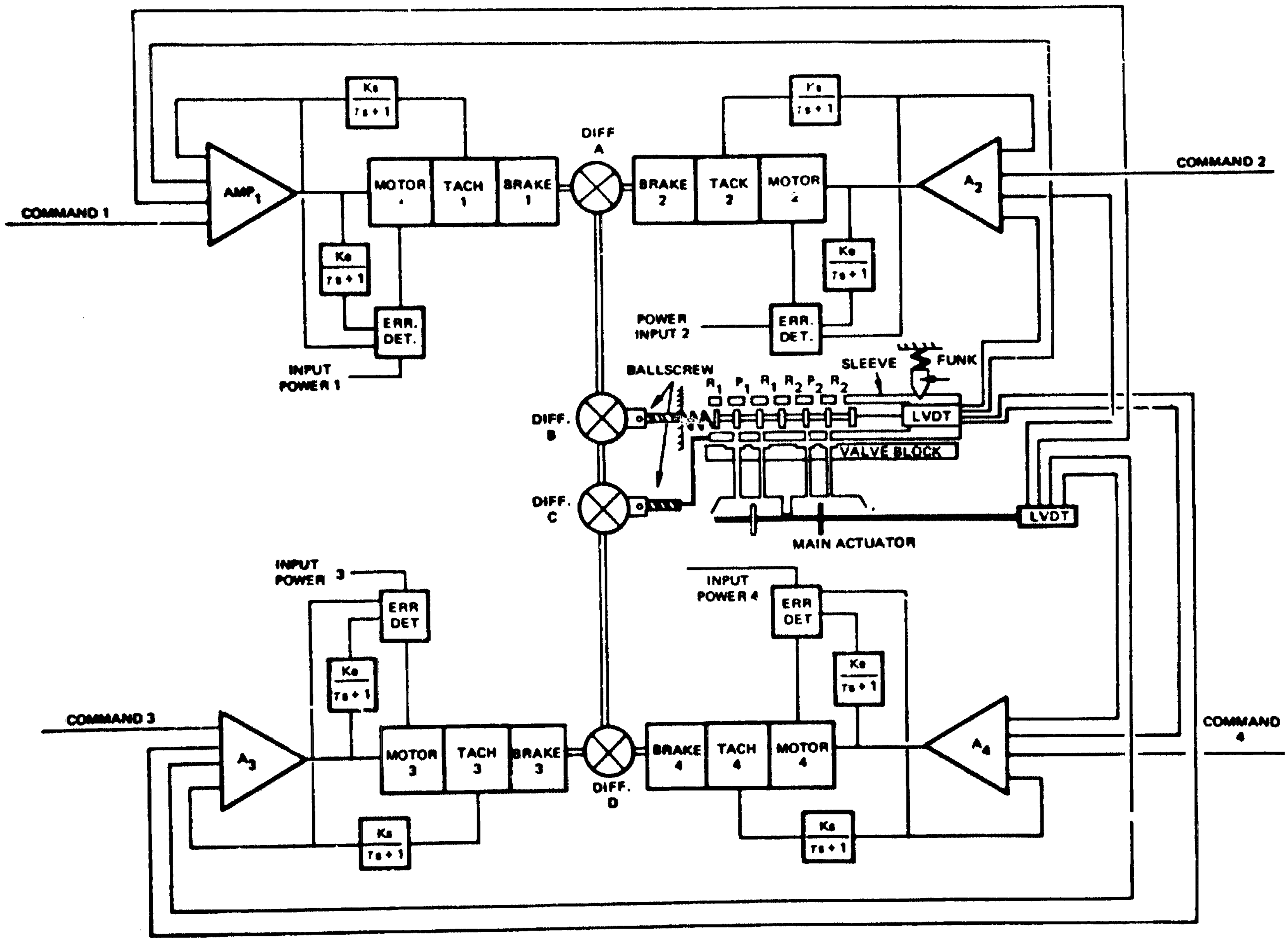

FIGURE A-51 DISPLACEMENT SUMMING WITH DIFFERENTIALS AND ELECTRICAL MOTORS INSIDE THE LOOP

CONCEPT 36-114 MODULAR ACTUATORS

This concept shown in Figure A-52 is based on developing basic functional elements as standardized "building-block" modules; designed to perform a basic function in the most efficient manner and capable of being utilized in conjunction with other modules for assembly of multiple actuator configurations; such as simple, parallel and tandem actuators with varying stroke capabilities.

These modules would be designed to provide maximum simplicity capable of being manufactured economically through use of simple machining techniques and need for only simple tooling.

The modular actuator concept would include a direct drive fly-by-wire control valve and a separate module for electrical feedback. Modules within the same size (load) actuator would be interchangeable through standardization, which will have significant impact on the maintenance and logistics system.

This concept then would provide approximately ten (10) basic modules, as currently conceived. These modules represent the essential "building-blocks" of all hydraulic actuators:

(a) Cylinder
(b) Piston rod
(c) Piston head
(d) Adapters #1 & #2
(e) Control Valve
(f) LVDT
(g) Connector plate
(h) End fitting
(i) Connecting tubes (porting)
(j) Retainers

The direct drive fly-by-wire control valve shown in Figure A-53 consists of a proportional control loop utilizing a 4-way spool and sleeve valve driven by a high output torque motor. The spool has two lands and is flow force compensated. Spool travel is 0.010 to 0.040 inches and spool overlap is 0.001 inches to insure null tracking over the entire operating temperature range. Flow forces will be kept below one pound. The torque motor will produce at least 40 lbs. drive force on the spool at null position. The torque motor will be kept dry to elminate accumulation of contamination at the permanent magnet. The spool centering springs will have a sufficiently high rate to insure positive centering at electrical zero and to provide sufficiently high frequency response. The valve housing will be compatible with modular dual actuators utilizing rip-stop design principles. The dual spools will have provisions for rigid mechanical synchronization. The entire valve, including housing, will be made of steel. The valve assembly will weigh less than 5 lbs. (each) or 10 lbs. per dual arrangement.

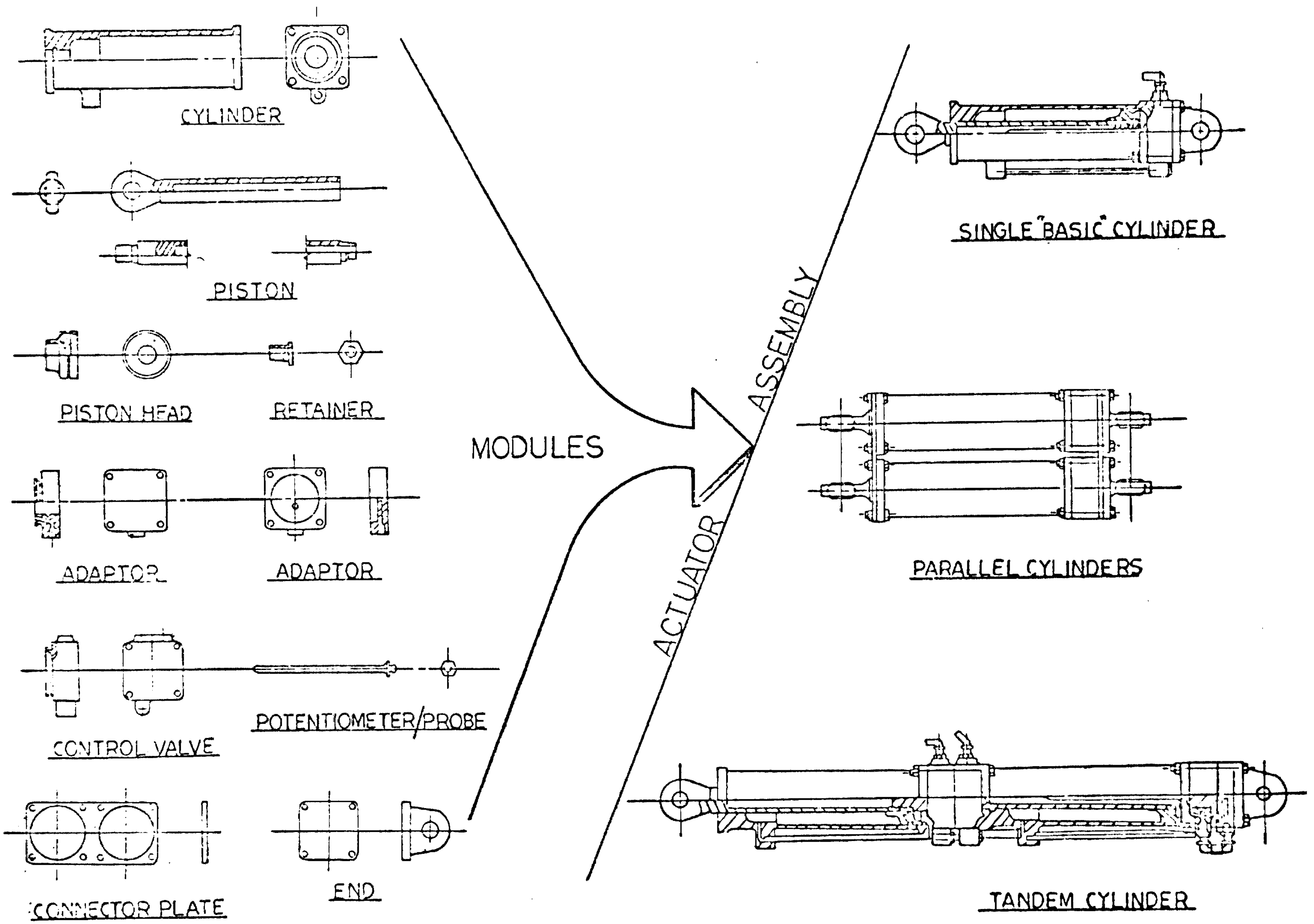

FIGURE A-52 MODULAR ACTUATOR CONCEPT

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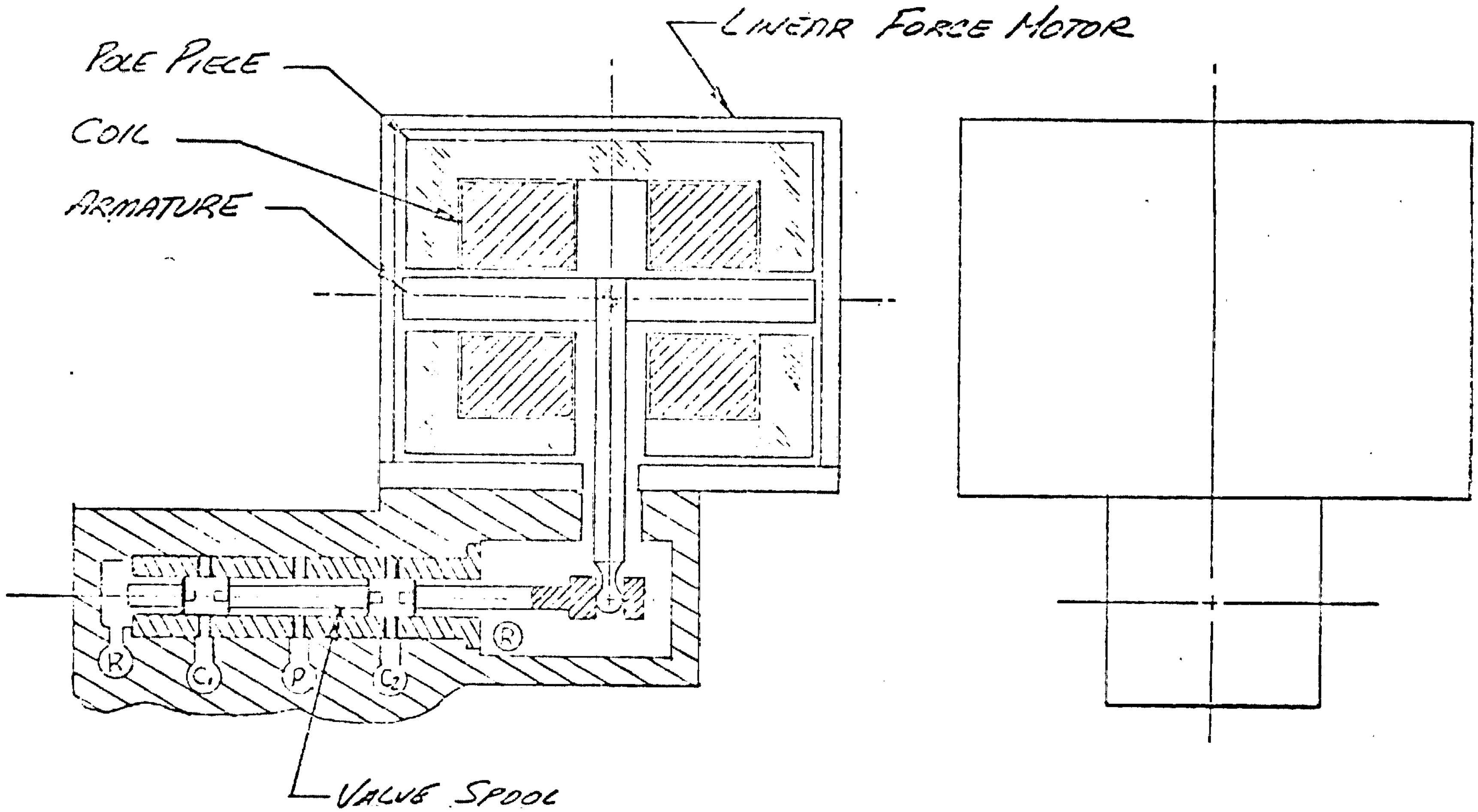

FIGURE A-53 DIRECT DRIVE CONTROL VALVE PACKAGING ARRANGEMENT

CONCEPT 37-005 B-52H ROLL CONTROL SYSTEM

Figure A-54 shows the fly-by-wire equivalent of the mechanical B-52H spoiler control system. Quadruplex position transducers at the control wheel provide roll command signals to the servovalves which drive the spoiler actuators. Mechanical feedback to the valves is employed as in the original system. Trim, AFCS, airbrake, and manual control signals are summed at the control electronics.

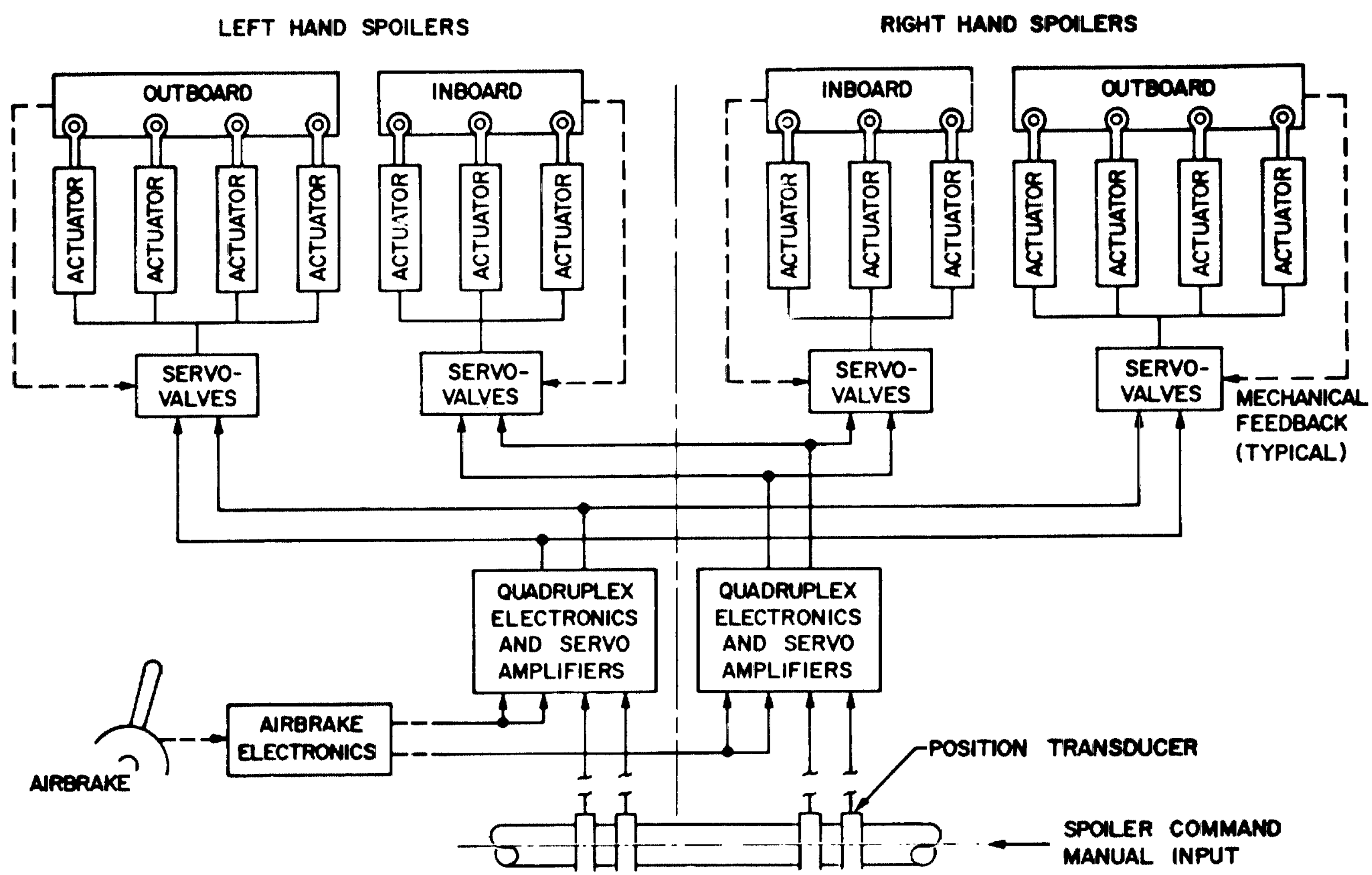

FIGURE A-54 B-52H ROLL CONTROL SYSTEM

CONCEPT 38 -009 SERVOACTUATOR

The double fault-correcting actuator, which remains undegraded after two faults, consists of three active channels, an electronic model and a comparator (Figure A-55). Each channel again is a complete independent actuator.

The pistons of all three channels are connected to the output shaft and the cylinders of the two outside channels are connected through a center-pivoted differential link attached to the main frame. The reaction loads of these two channels appear at this joint. In normal operation, the center cylinder remains disengaged and transmits no load.

The piston position in each channel is fed back both to the channel's servo amplifier and to the comparator, which detects a failed channel by comparing the three position voltages and the model output voltage. Its logic networks have a built-in threshold that makes them insensitive to the normal tolerance variations of these signals. When one of the four voltages indicates a position difference in excess of tolerance, however, the logic deactivates the failed channel. A signal from the comparator removes electric power from the shutoff valve in the hydraulic supply to the channel in question. If this channel is on the outside, the result is that, on its side of the actuator, the differential link is locked by spring action and a spring-loaded bypass valve opens to let oil circulate freely around the piston of the channel. The other outside channel now can drive the load with undegraded performance, since the differential link has been locked to the main frame and the piston in the failed channel is free in its cylinder.

The logic networks continue to compare the remaining two channels and the model. Should a fault occur in the remaining outside channel, the comparator deactivates this channel and, by removing the pressure from the spring-loaded center link lock, engages the center cylinder.

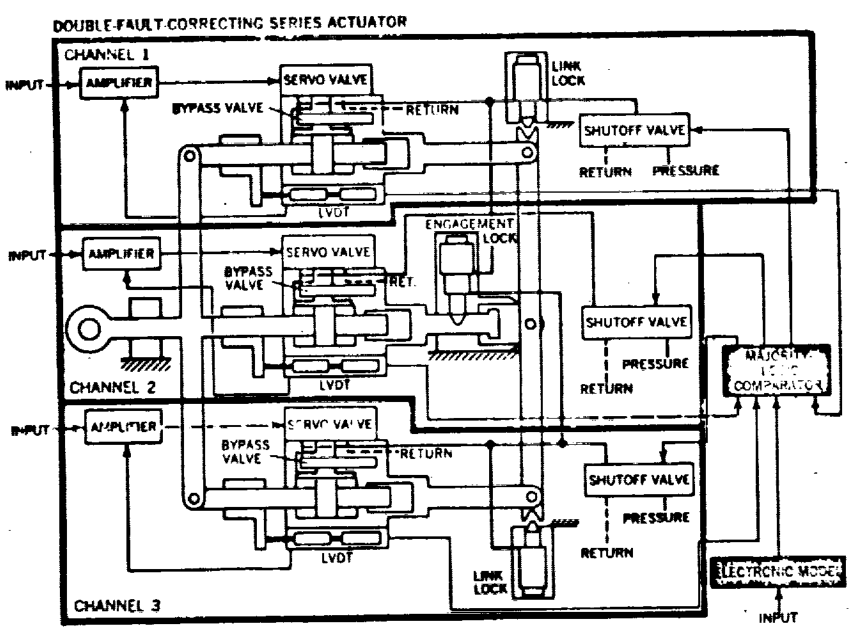

FIGURE A-55

CONCEPT 39-005 TRIPLE PARALLEL HYBRID SERVO SYSTEM

Position-Monitored Standby Redundant Actuator configuration employs three parallel position servo-actuators with common outputs and an electronic model and monitoring (Figure A-56). Each servo channel consists of an actuator, two-stage servovalve, servo amplifier, and dual LVDT position transducers. The actuators are tied together rigidly so that relative motion does not occur. The outer channel cylinder bodies are connected through a differential link pivoted at the center where it is attached to the main actuator body. The outer channels are normally active. The center actuator is normally in standby; it engages only if both outer channels fail. This is a hybrid active/standby type of redundancy.

Operation of the outer channels (A and B) is normally like a position summed actuator. The differential link motion is small (caused by tolerance variations) if the A and B outputs are equal. Output motion equals the average output of A and B. On failure of either A or B, the failed actuator is bypassed and its link end is locked. The outer channel then supplies the output with undegraded performance. Failure of the second channel causes its actuator to be bypassed, its end link to be locked, and the center actuator (C) to be engaged. Channel C then drives the load with undegraded performance. A third failure centers and locks the actuator. The locks, bypass valves, and engage valves operate on command from the electronic monitor via electrohydraulic solenoids. The monitor operates on the actuator position transducer outputs and the model output.

The configuration has the following advantages:

(1) No performance degradation after failures.

(2) Failure isolation is maintained.

(3) Mechanization expandable to higher redundancy.

(4) Servovalve spool position transducers not required.

The configuration has the following disadvantages:

(1) Output deviation required for failure detection.

(2) Depends on solenoid valve, bypass valve, and lock and monitor reliability for transfer.

(3) Fast transfer times require high speed solenoids and monitor to minimize transients.

(4) Monitor sensitive to large load variations because of the standby channels.

CONCEPT 3^9-005 (continued)

(5) Monitoring may be sensitive to large power transients.

(6) Increased size and weight because each actuator must be sized to carry the load.

Comments:

By using the output as a small secondary actuator to mechanically drive a power actuator valve, disadvantages 1, 4, and 6 could be minimized or eliminated.

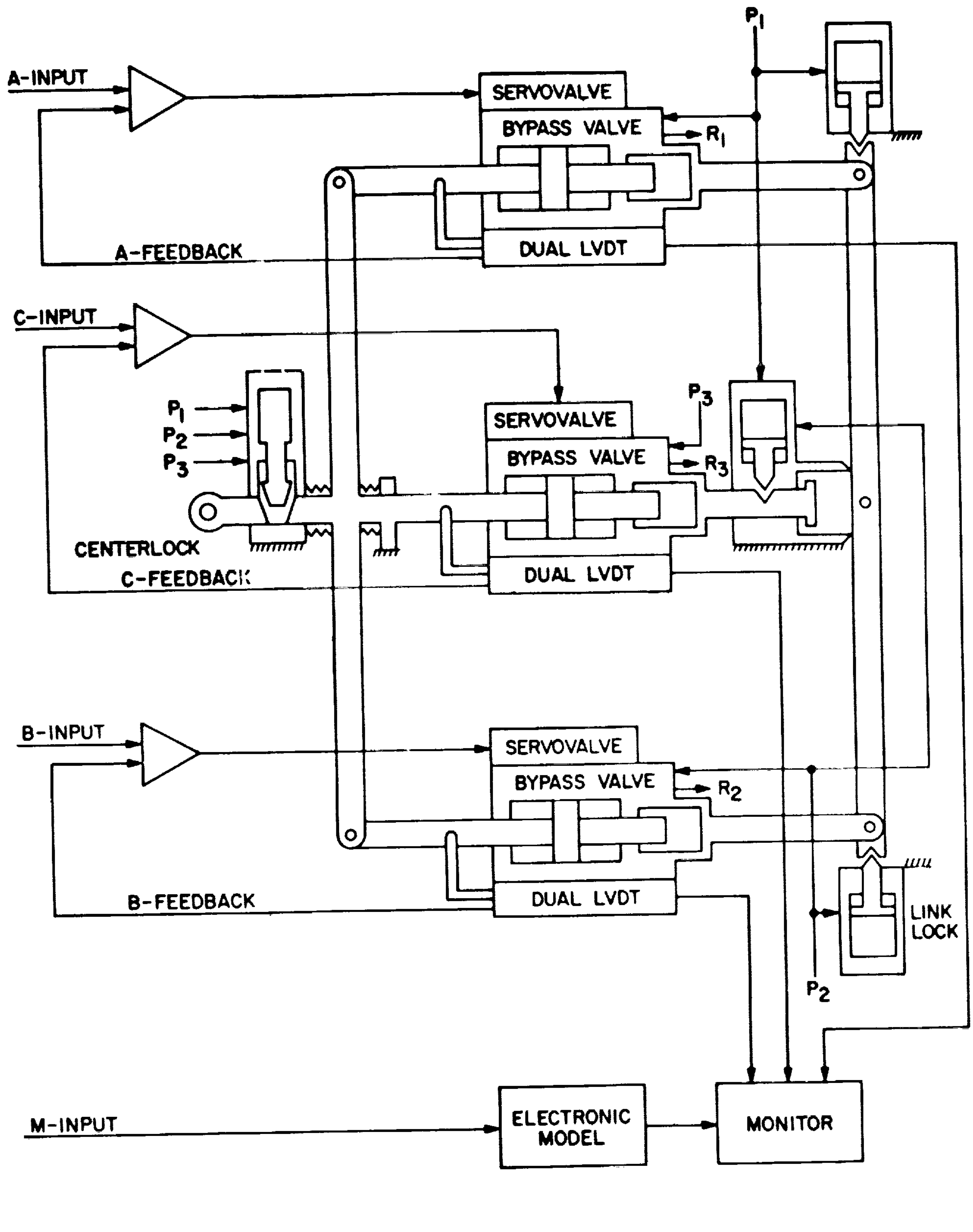

FIGURE A-56 POSITION MONITORED STANDBY REDUNDANT ACTUATOR

CONCEPT 40-060 F-8 DIGITAL FLY-BY-WIRE SYSTEM

The modifications to the F-8 airplane are shown schematically in Figure A-57. Secondary actuators were installed in series with the basic F-8 primary actuators. Five actuators were required, one for the rudder and one each for the two horizontal stabilizers and the two ailerons. These secondary actuators are driven electrically by either the primary digital system, which consists of Apollo guidance system hardware, or by a triplex analog backup system. Most of the primary digital system is on a removable pallet behind the cockpit. The backup system electronics are in the lower left fuselage bay.

The overall system mechanization is illustrated in Figure A-58. The pilot's stick , pedal, and trim inputs are routed to the digital computer as inputs to the primary system. As the central element in the system, the Apollo LGC must interface with the actuators and the IMU. The subsystem, consisting of the LGC, the IMU, and a coupling data unit (CDU) which contains the interface digital to analog (D/A) and analog to digital (A/D) converters, is taken directly from the Apollo guidance system, thus maintaining the high level of integrity characteristic of Apollo equipment.

A single channel digital primary configuration was chosen for the first flight phase for experience with the digital control aspects early in the total fly-by-wire program. The total system design called for two-fail operational reliability; therefore a triplex backup system was necessary. Protection against failure in the analog portion of the primary channel was provided by dualizing the downstream paths.

The two drive paths feed the active and monitor servo-valves. If a failure occurs in either the active or the monitor path, a hydraulic comparator senses a differential pressure between the active and monitor servovalve and transfers to the backup control system (BCS) using hydro-mechanical switching. As long as the system is in the primary digital channel, the backup electronics track the active channel by way of the sync network. Only the hydraulic pressure is bypassed at the secondary actuator, keeping the BCS ready to take over at any time. If a transfer to BCS is requested, the bypass is removed and the sync network is disabled, resulting in immediate proportional control from the pilot's stick. While in the BCS mode, the active serovalve is bypassed and the secondary actuator operates as a force summer for the three backup channels. The digital computer continues to operate, computing the control laws which give the best estimate of what the BCS commands. If a transfer to primary channel is attempted, the transient should be small, assuming the computer was tracking the BCS. If the primary channel fails to track accurately, a cross-channel comparator (not shown in the figure) prevents transfer to the primary channel should the error between the active channel and the BCS be excessive.

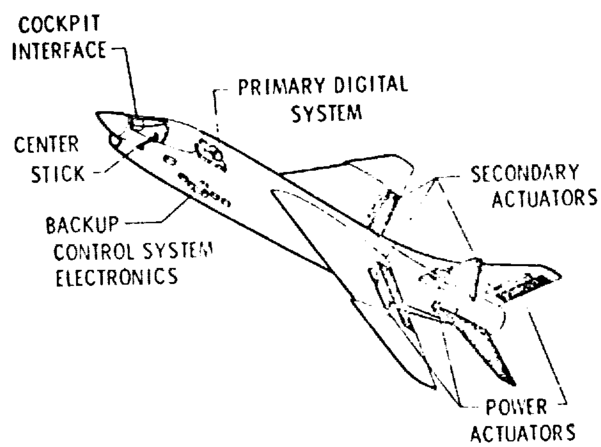

FIGURE A-57 F-8 AIRPLANE WITH DIGITAL FLY-BY-WIRE MODIFICATIONS

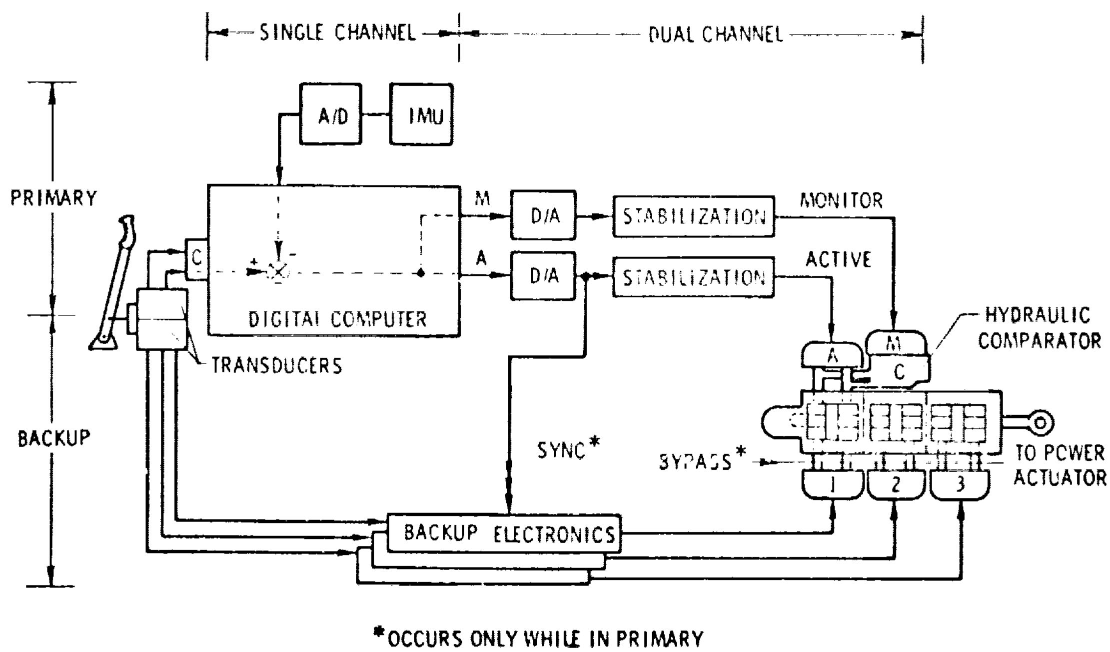

FIGURE A-58 F-8 DIGITAL FLY-BY-WIRE SYSTEM MECHANIZATION

CONCEPT 41-051 SPACE SHUTTLE SERVOACTUATOR

This servo actuator is an electrohydraulic, three-channel, active-standby configuration developed by Hydraulic Research and Manufacturing Company (HRM). This two-fail/operate, fail/passive actuator is an implementation of redundant hydraulic control employing an intrasystem monitoring design.

A modular design approach is used to provide the required redundancy. This actuator consists of three independent systems or modules with complete hydraulic isolation controlling a triple tandem piston. Only one system controls the actuator at any one time. With a malfunction in the controlling system, a switch is made to a standby system, thus, there is no loss in output force or performance degradation.

Referring to Figure A-59 , after hydraulic pressure is applied, the three solenoid valves are pulsed to engage the system. Once pulsed, the solenoid valve is held on the seat with system hydraulic pressure. This pressure drives the three engage valves against the engage valve spring and activates System #1. The active servo valve in System #1 controls the actuator.

The servovalve consists of an electrical torque motor and hydraulic output stage. The output stage of this two-stage valve is closed center, which means that the spool is designed to block fluid flow when at the null position. Current flowing in the torque motor coils induces a torque in the armature, which pivots the flapper slightly toward either nozzle. This motion unbalances the hydraulic amplifier circuit, causing a pressure difference to be generated between the two end chambers of the second stage spool. This pressure difference creates motion in the second stage spool which varies the flow metering area in the sleeve, thus, changing the output flow.

Flow proportional to input current is achieved by the use of rectangular metering slots, and by restraining the spool with a feedback spring referenced directly to the torque motor armature. This mechanical feedback feature produces a torque on the armature proportional to spool displacement. The torque transmitted to the armature by the feedback spring opposes the torque induced by the input current. Equilibrium balance of the torques results in spool displacement and flow proportional to input current.

The valve is modified with the addition of a second stage monitor flapper nozzle (see Figure A-59). The function of the monitor flapper nozzle is to develop pressures proportional to the position of the second stage spools of the active and monitor valves. These two pressures are fed to opposite ends of the comparator spool. If no malfunction occurs these two pressures will vary but will remain equal in magnitude and the comparator spool will remain centered.

If a malfunction occurs, the outputs of the active and monitor valves will differ. This will cause a pressure difference on the comparator spool creating motion of the spool. When the pressure difference exceeds a predetermined threshold, motion of the comparator spool will dump the supply pressure holding the engage valve to return. The engage valve of System #1 will be forced by the spring pressure into a bypass position. The bypass position blocks the output of the active servovalve of System #1. The engage valve of System #2 will move one step and System #2 will become the active channel and will operate in exactly the same way as System #1.

The failure threshold of the comparator can be easily varied by spring rate on, and overlap of, the comparator spool. Once the optimum threshold is determined it will remain fixed in design.

If a malfunction occurs in System #2, a switchover to System #3 will be accomplished in the same manner. If System #2 has previously failed, the switch will be from System #1 to System #3. The sequence of system failure is no problem. In this design, only a channel that is operational is capable of gaining control of the actuator.

A third failure will cause the actuator to fail in a bypass mode on all three systems. System failure is detected by a pressure switch on the comparator valve.

Pressure loss, exceeding a predetermined threshold in any system, will cause the ball in the solenoid valve to unseat, thus, switching to the next channel.

After malfunction, any one system will not come back on line until the solenoid valve is pulsed. If the malfunction has been corrected, pressure will hold the solenoid valve ball on its seat, input to the comparator spool from the active and monitor valve will be identical, the engage valve will be pressurized, and the pressure switch will cycle, thus, returning the system to normal operation. If the malfunction is still present, the system will immediately switch off line as before.

Attached to the actuator output are four position feedback linear variable differential transducers (LVDT's) (Figure A-60). One LVDT is dedicated to each of the three channels for servo stabilization and all four LVDT signals are sent to a failure detection logic. This logic uses a cross-channel failure detection method. Each LVDT signal is compared with the signals from all other working LVDT's. A fail decision is made if the signal of that LVDT differs appreciably from that of the other LVDT's. The failure threshold is an error voltage equal to that generated by displacing the actuator five percent of full travel. The detection of a failure energizes a latching relay which provides a positive d.c. bias voltage to the monitor servo amplifier. This causes the hydraulic logic to disengage the failed channel.

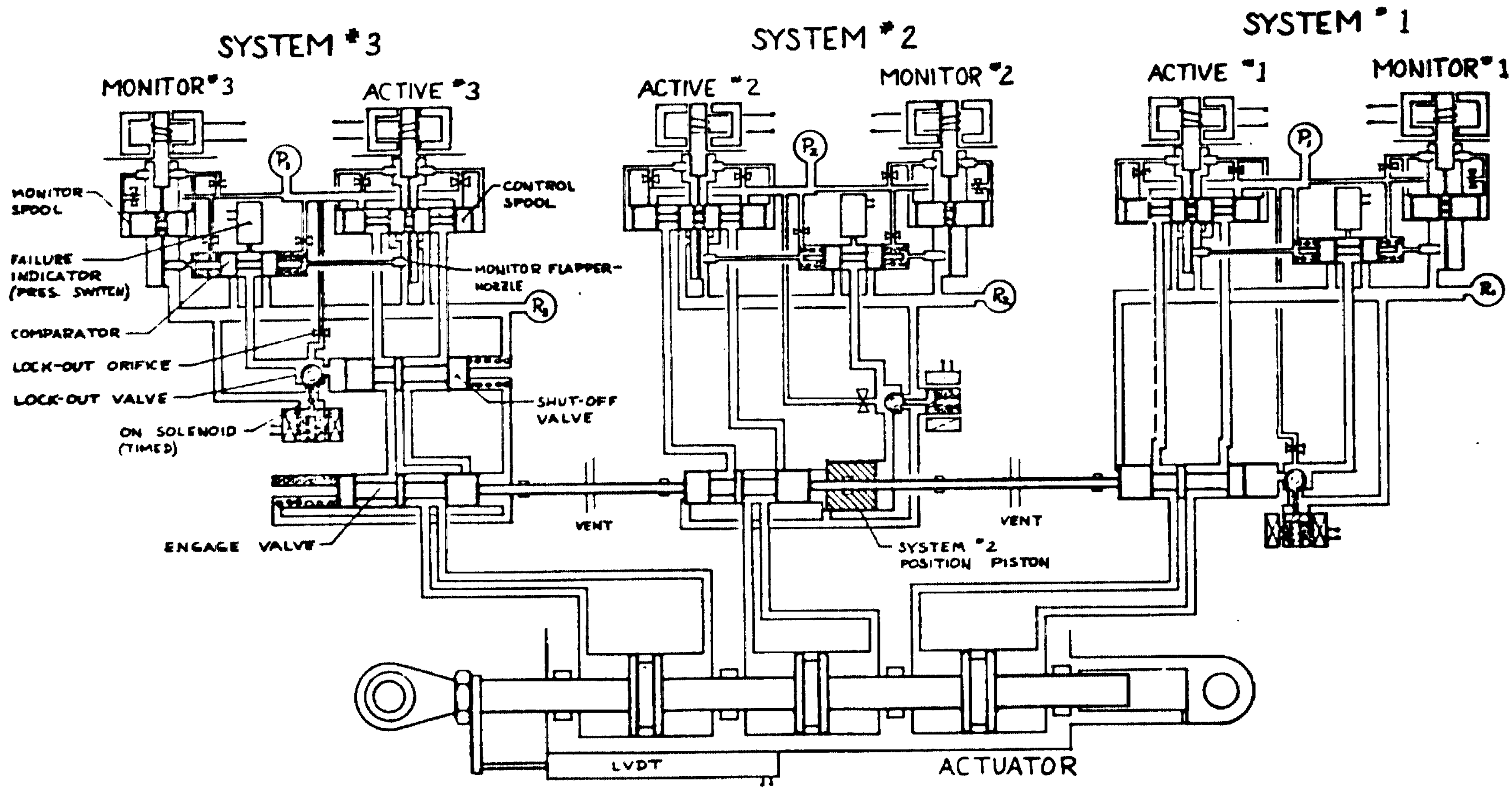

FIGURE A-59 HYDRAULIC SCHEMATIC - ARM A

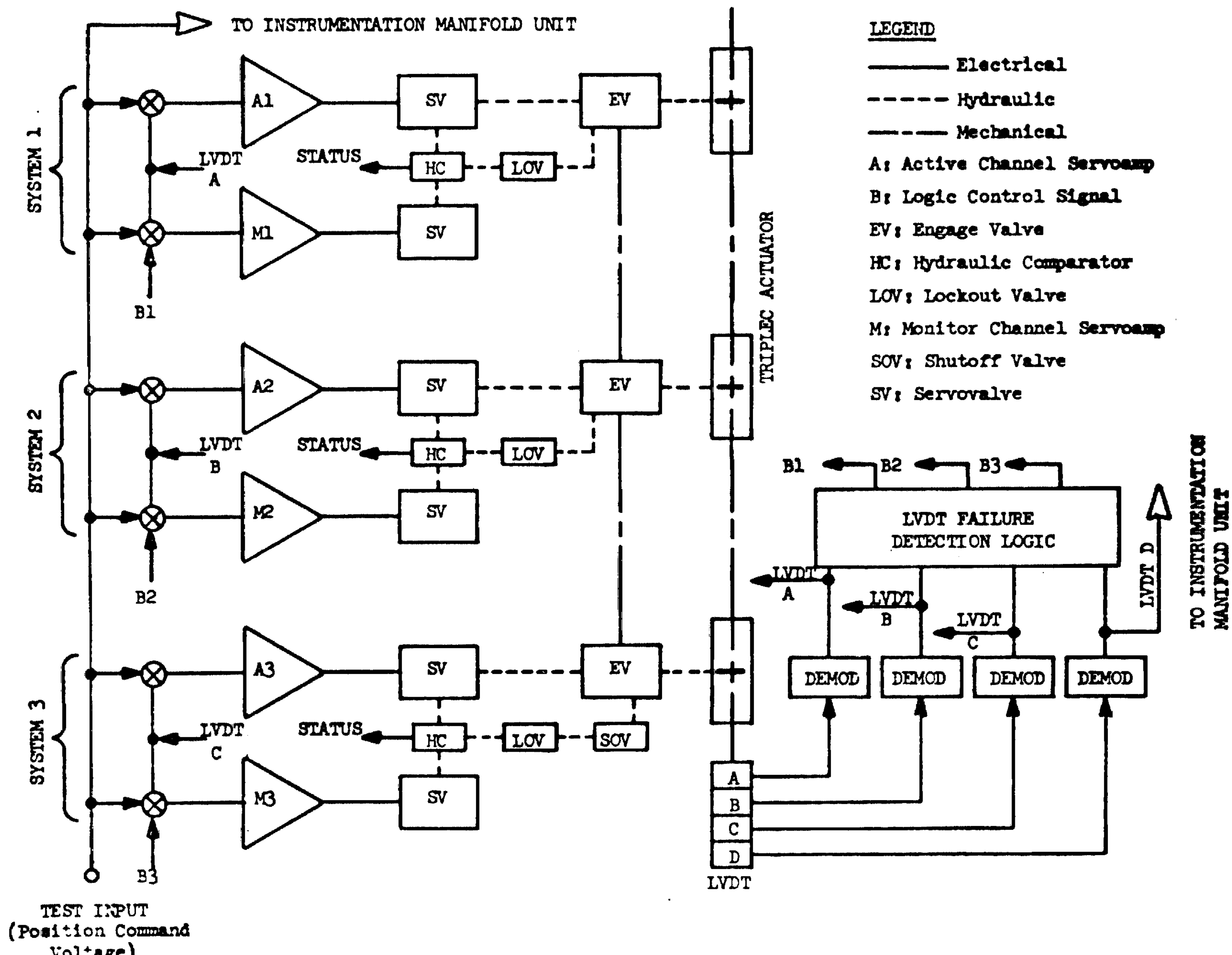

FIGURE A-60 ELECTRICAL SCHEMATIC - ARM A

Secondary Actuator with standby redundancy configuration (Figures A-61 and A-62) has a small redundant secondary servoactuator which mechanically drives the main control valve and power actuator with nearly unity feedback. The dual feedback linkage can be sealed within the actuator body where it is protected and bathed in oil. The power actuator employs active redundancy. The secondary servo uses three real channels (identical within tolerances) and a model channel which may be hydraulic or electronic. One real channel is active while the others operate in standby by driving dummy model pistons which are sized to match the secondary actuator. The feedback of the model piston and the secondary actuator is electrical. The servo valves are coupled to the secondary actuator through a four-position engage valve. The engage valve transfers the system through its operational modes on commands from the electronic monitor via electrohydraulic solenoids. The monitor compares the position of the secondary actuator and model pistons. This eliminates the need for servovalve spool transducers and allows limited monitoring of the main control valve and power actuator. Complete channel isolation is maintained. Failures are detected without requiring power actuator motion from the commanded position although some motion will occur for hardover failures.

For the first failure, the system switches from the active secondary channel to a standby channel unless the failure is in a standby channel. In this case, the monitor prevents that channel from ever being engaged. For the second failure, the system switches to the remaining standby channel. Channel switching proceeds as follows assuming an active channel failure. The engage valve disconnects the active valve from the actuator and bypasses the actuator while simultaneously switching the standby valve from its model piston to the secondary actuator. Also simultaneously, the monitor switches the feedback of the standby channel from its model piston to the standby feedback transducer on the secondary actuator.

The configuration has the following advantages:

(1) No performance degradation due to failures
(2) Failure isolation is maintained (Figure A-62 only)
(3) Monitor insensitive to load variations
(4) Mechanization easily expanded to higher redundancy
(5) Actuator deviation not required for failure detection
(6) Servovalve spool position transducers not required
(7) Small servovalves are adequate
(8) Power servo channels are active which minimizes size

The configuration has the following disadvantages:

(1) Depends on solenoid valve and engage valve reliability for transfer
(2) Fast transfer times require high speed solenoids and comparators to minimize transients
(3) Monitor may be sensitive to large power transients
(4) Part of secondary failure transient is transmitted to the output through the mechanical linkage
(5) Requires a secondary actuator
(6) Requires model pistons and/or an electronic model

CONCEPT 42-005 - (continued)

Comments:

This configuration could be designed for active redundancy using synchronization to eliminate disadvantages 2 and 4 but at the expense of added complexity.

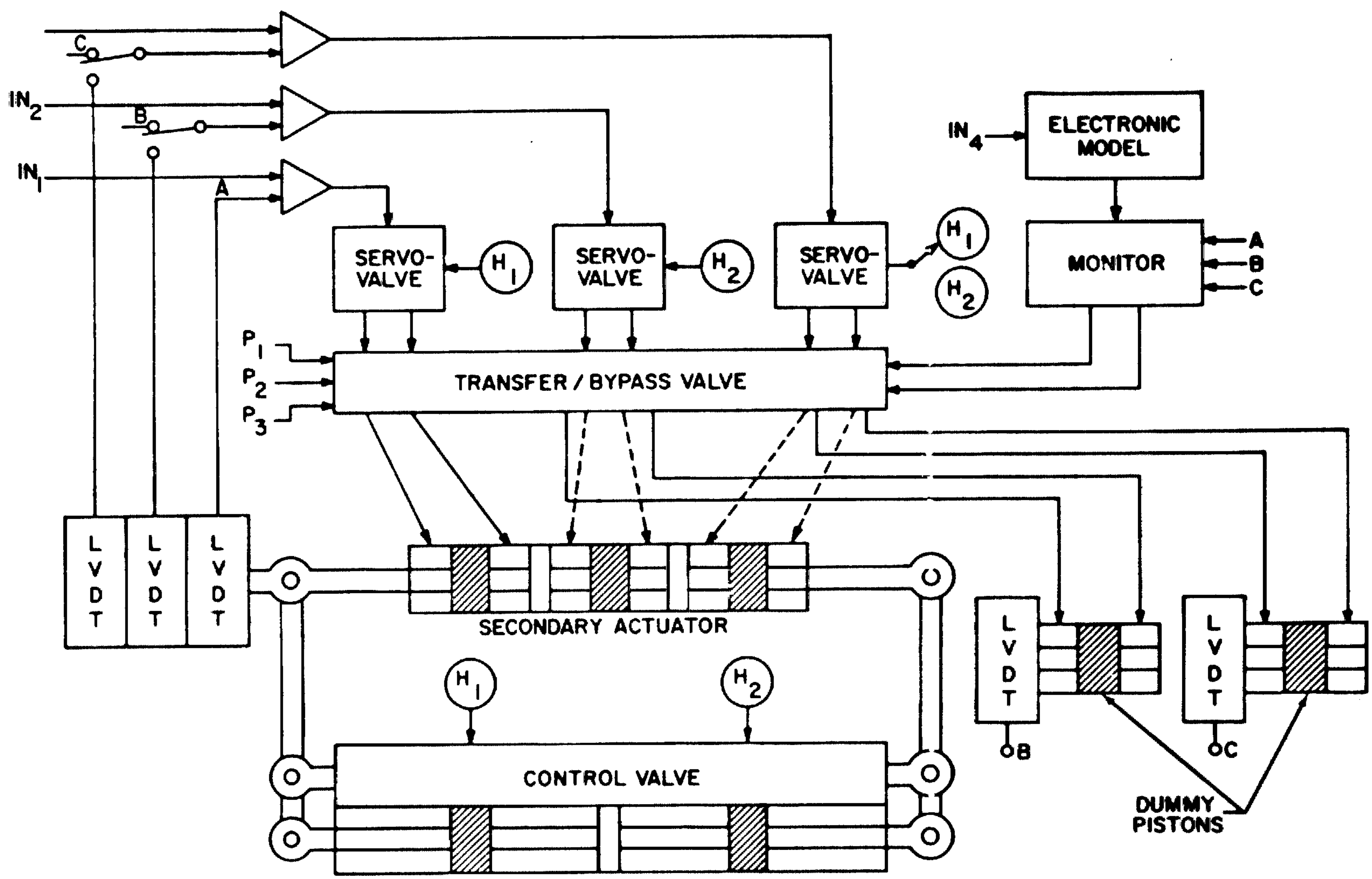

FIGURE A-61 SECONDARY ACTUATOR WITH STANDBY REDUNDANCY

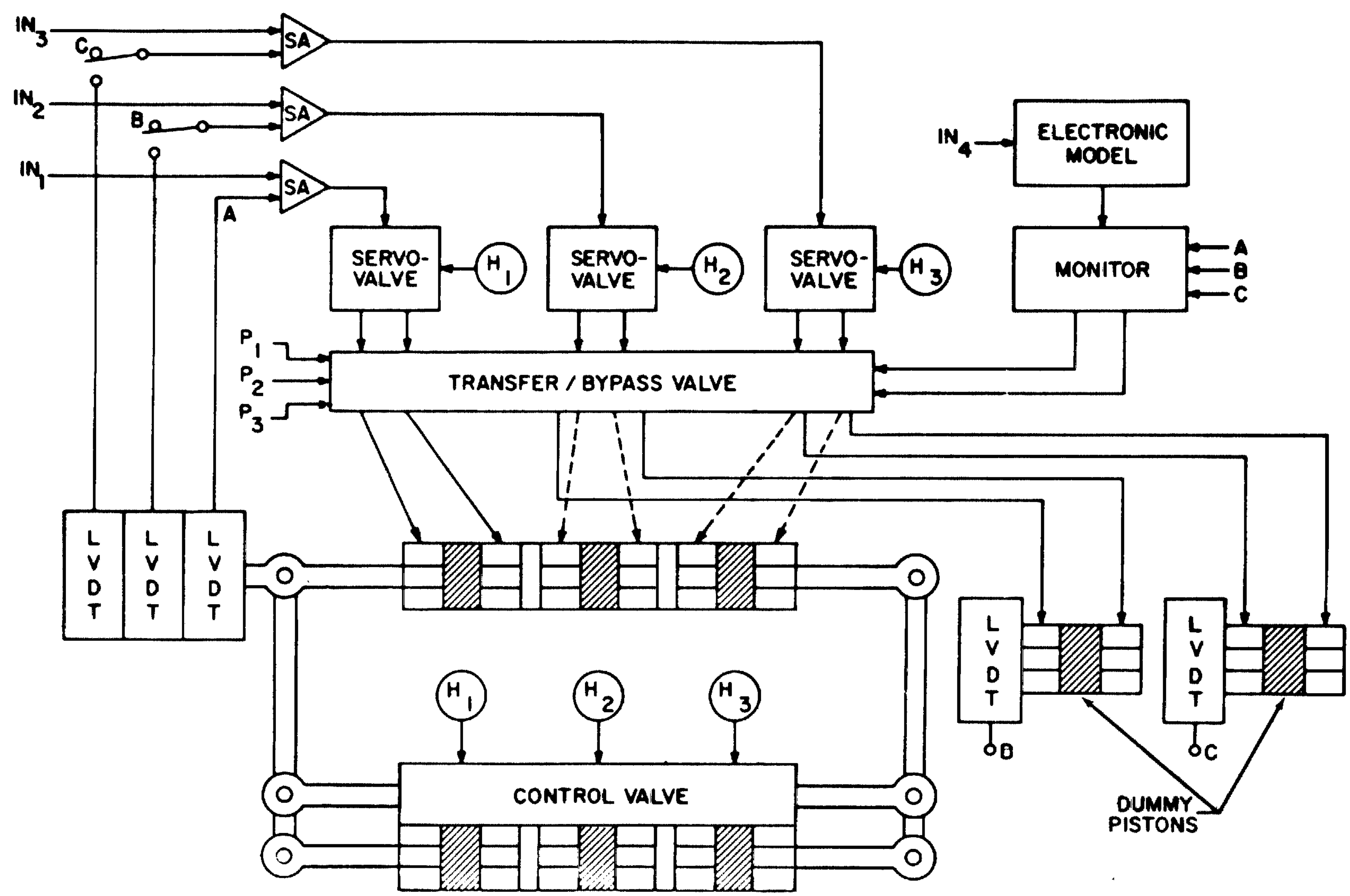

FIGURE A-62 SECONDARY ACTUATOR WITH ISOLATION MAINTAINED

CONCEPT 43-005 QUAD PARALLEL ACTUATOR SYSTEM

The Position-Monitored (Hydraulic) Standby Redundant Actuator configuration shown in Figure A-63 employs four parallel position servoactuators whose outputs are rigidly connected so that relative motion does not occur. One channel (A) is active, two channels (B and C) are in standby, and one channel (D) is a model. A channel consists of an actuator, two-stage servovalve, servo amplifier, and LVDT position transducer. A series of engage valves and locks connect the various channels to the load, one at a time, on command from the hydraulic monitor. The actuator cylinders are sleeves that move within the main actuator body against a centering spring load except channel A which is the reference channel whose sleeve is fixed. Porting between the sleeves and the body provides hydraulic position comparison with the reference channel for voting. Under normal conditions no relative motion occurs. Upon failure of channel A, relative motion occurs in all three sleeves. Channel A is bypassed and the sleeve of channel B is locked to the body thus engaging it to the load and making it the new reference channel. If any other channel fails first, its sleeve alone moves. The resulting vote causes the actuator of that channel to be bypassed. A third failure causes center lock because of the disagreement of sleeve positions.

The configuration has the following advantages:

(1) No performance degradation after failures.

(2) Failure isolation maintained.

(3) Mechanization easily expanded to higher redundancy.

(4) Servovalve spool transducers not required.

(5) Transfer is fast because electrohydraulic solenoids are not used.

The configuration has the following disadvantages:

(1) Output deviation required for failure detection.

(2) Depends on engage valve and lock reliability for transfer.

(3) Silting may affect hydraulic comparator performance by increasing threshold.

(4) Increased size and weight because each actuator must be sized to carry the load.

(5) Monitor sensitive to large load variations because of the standby channels.

Comments:

By using the output as a small secondary actuator to mechanically drive a power actuator valve, disadvantages 1, 4, and 5 can be minimized or eliminated.

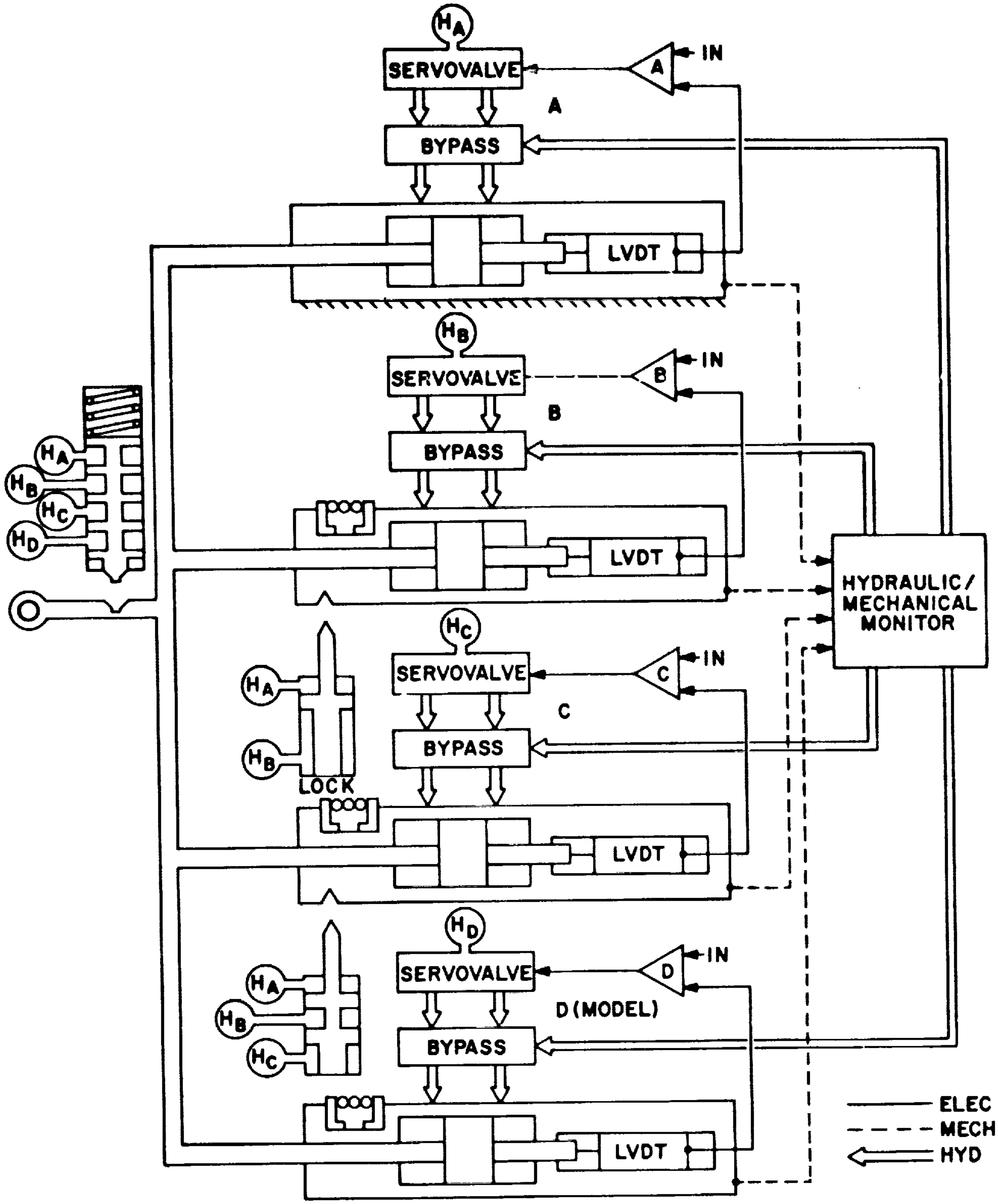

FIGURE A-63 POSITION MONITORED (HYDRAULIC) QUAD PARALLEL ACTUATOR

CONCEPT 44-131 SPACE SHUTTLE REDUNDANT FLIGHT CONTROL

This redundant minilogic servo actuator system (Figure A-64) was developed by the Hydraulic Research and Manufacturing Company of Valencia, California, for the NASA space shuttle TVC servo actuator. The system is a two channel electro-hydraulic, with a hydraulic comparator (hydromechanical), self-monitoring (failure detection), active-standby and a two fail operate-fail safe (trail) capability. System control is accomplished by one active and one monitor two stage electro-hydraulic servovalves with monitor provisions per electrical channel. The main actuator piston is a dual tandum type with an area of .5199 square inches and a total stroke of 2.00 inches.

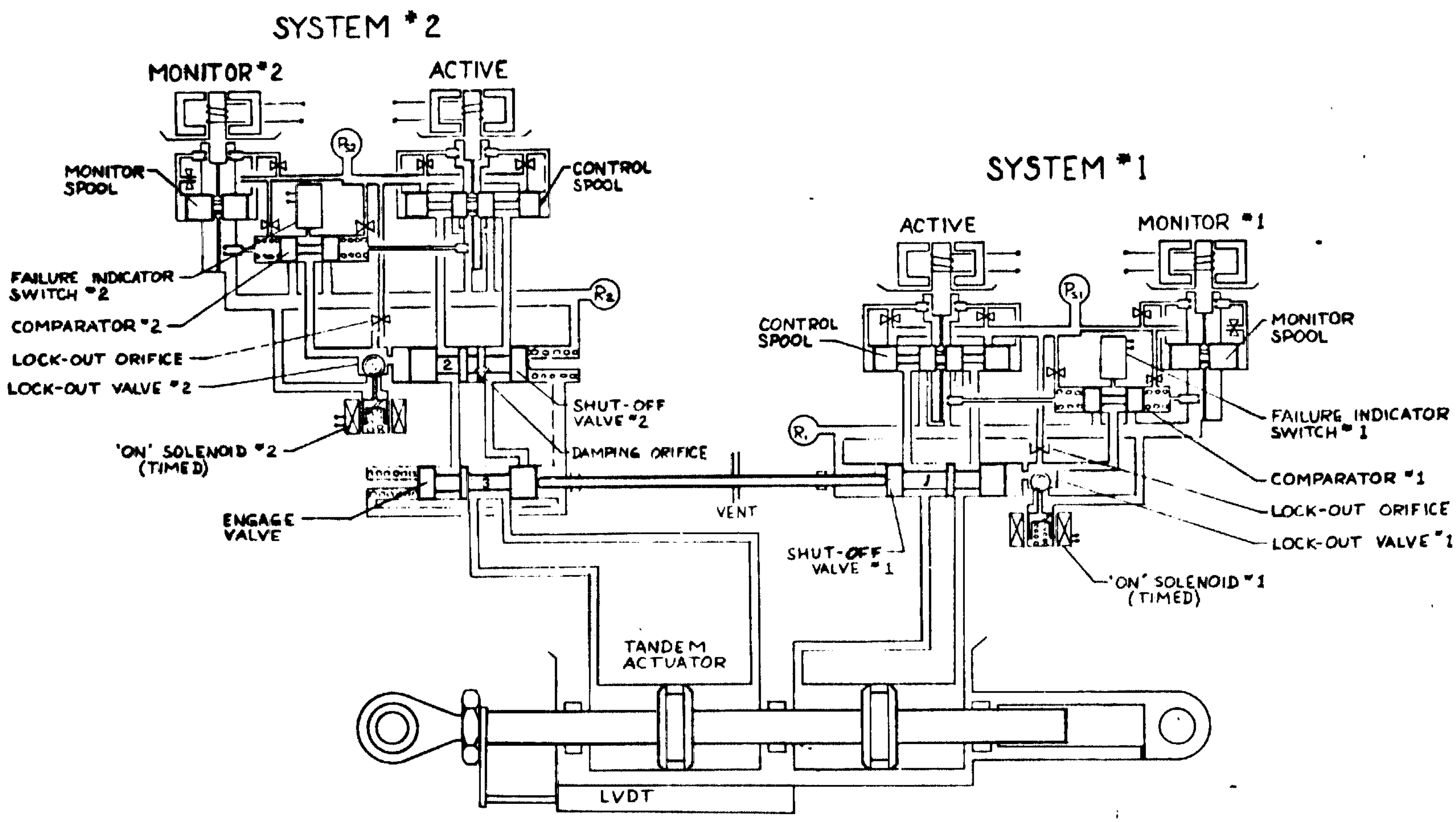

FIGURE A-64 SPACE SHUTTLE REDUNDANT FLIGHT CONTROL UNIT

CONCEPT 45-092 ACTIVE STANDBY WITH ELECTROMECHANICAL ACTUATOR INSIDE THE FEEDBACK LOOP 680J

The main dual tandem servo valve of this system is driven by electromechanical actuator number 1 through a solenoid operated clutch, Figure A-65. All E/M actuators operate together, but clutches number 2 and number 3 are not engaged. A monitor actuator, number 4 is used to provide an additional position signal for comparison with the other three. Main actuator position is fed back to each E/M actuator by means of a 4-channel LVDT.

The position feedback signal of each E/M control actuator is amplified in its respective monitoring channel for comparison in the following combinations: 1-2, 1-3, 1-4, and 2-4. Should actuator number 1 lose power, jam, or fail in any way, the difference in its position feedback signal relative to those of actuators 2, 3, and 4 would cause the monitoring channels to trigger and lock comparator relays 1-2, 1-3 and 1-4. Relays 1-3 and 1-4 complete the circuit for triggering relay B which in turn switches power from clutch number 1 to clutch number 2. Relays A and B are wired such that they are self-locking and cannot be released except by means of a circuit breaking reset button. If actuator number 2 fails, relay A is triggered and power is switched to clutch number 3.

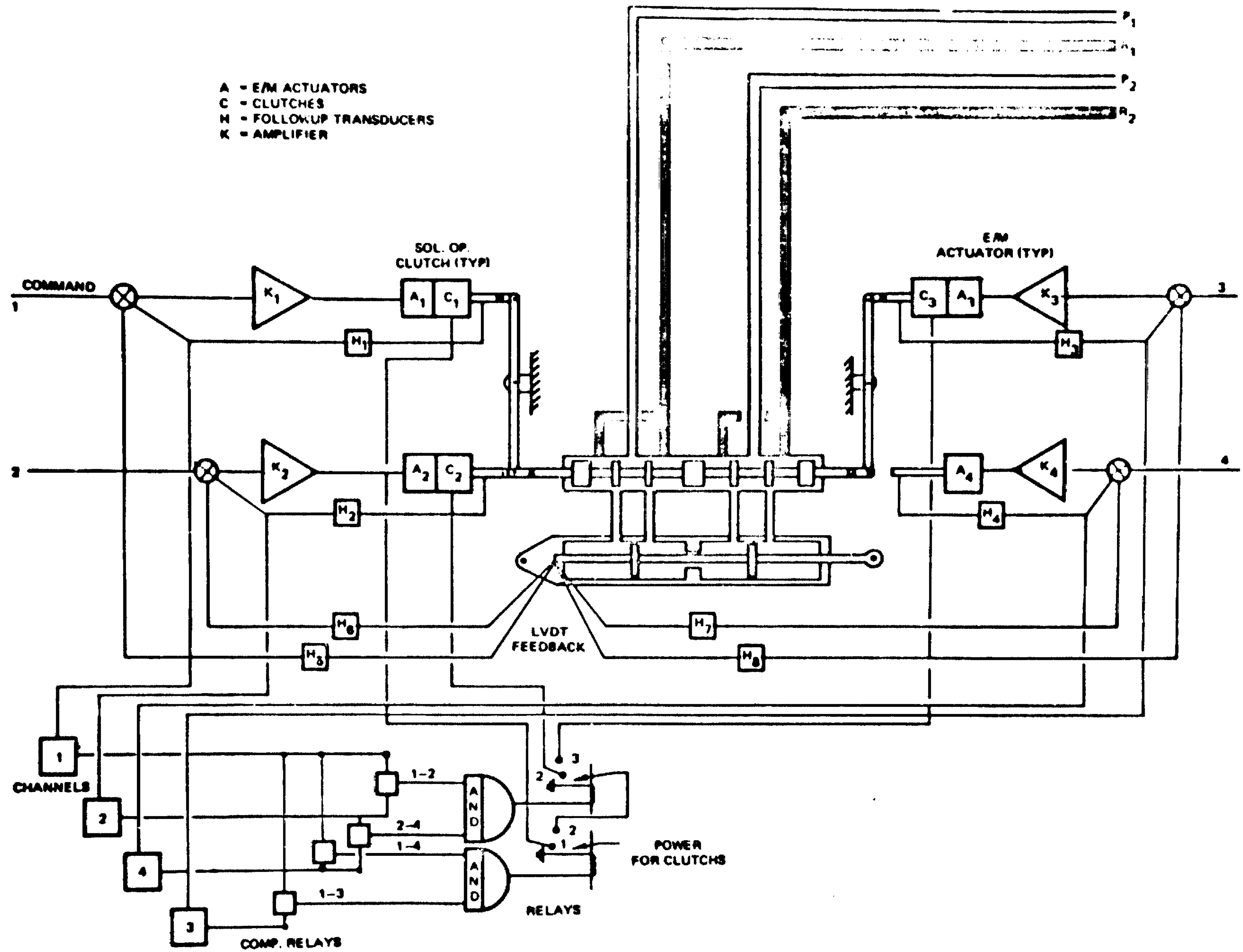

FIGURE A-65 ACTIVE STANDBY WITH ELECTROMECHANICAL ACTUATOR INSIDE THE FEEDBACK LOOP

CONCEPT 46-092 ACTIVE STANDBY WITH ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP 680J

The operation of this system is essentially identical to that of Concept 45-092 except that the position feedback from the main actuator is mechanical, not electrical, and is summed down stream of the signal converters, Figure A-66. The E/M actuator selection system is the same as that of Concept 45-092, eliminates hydraulics in the logic, and tolerates two signal failures plus one hydraulic failure.

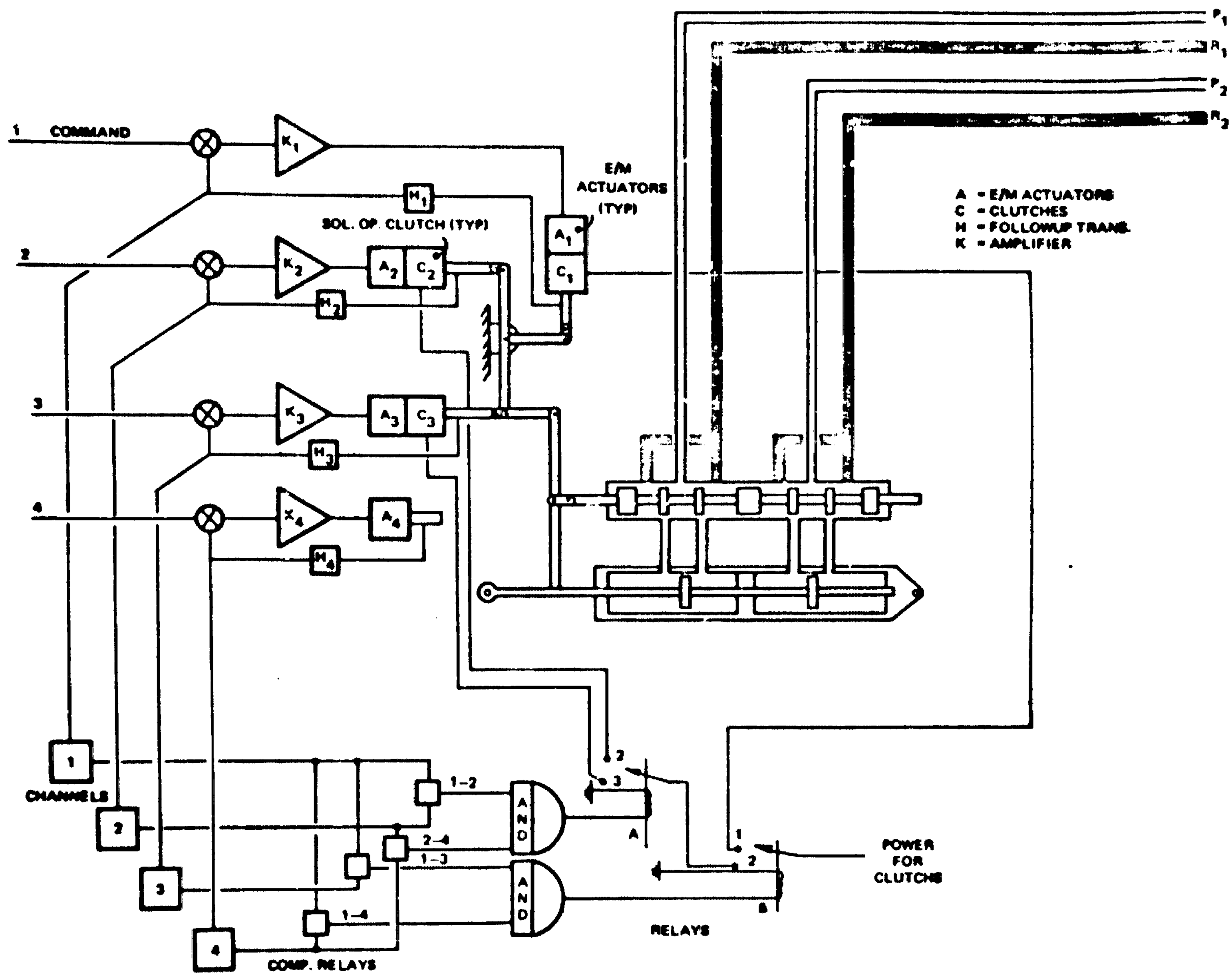

FIGURE A-66 ACTIVE STANDBY WITH ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP

The servoactuator package, which is shown in Figure A-67, is a completely self-contained, integrally-redundant, all hydromechanical unit. The package is supplied with three simultaneous electrical commands, together with dual hydraulic system pressurization. The unit contains self-error-sensing provisions so that the output rod continues to be positioned proportional to input command following failure of any input or of either hydraulic supply.

The basic redundant configuration can be classed "detection-correction" as opposed to "majority voting". In the broad sense, this means that normal operation is controlled by either one or two of the input commands, while the third input provides a reference or model function. Comparisons are then made between pairs of the inputs to detect the degree of mismatch which may exist. An allowable "working range" of mismatch is provided for anticipated drifts, gain mismatch, and other uncertainties. Whenever this working range is exceeded, an error is presumed and corrective action takes place. The correction may consist of switching-off the failed channel, or changing-over from one channel to a standby channel. If full failure protection is required then an indication of failure in either the reference or standby channels is necessary. The error comparisons in a detection-correction redundant configuration can be performed in many different ways and at many different locations in the system, with attendant advantages and disadvantages.

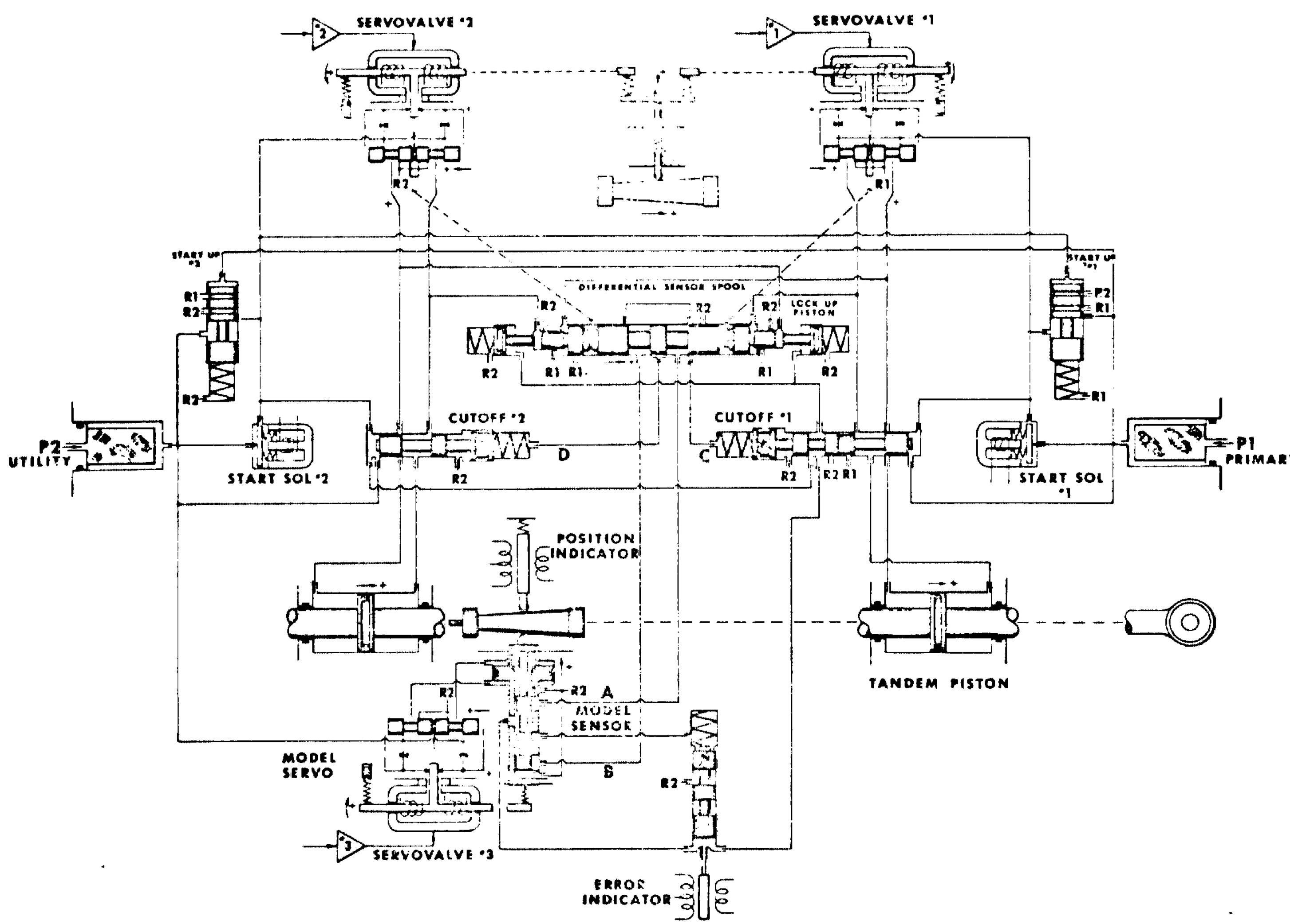

FIGURE A-67 SCHEMATIC F-111 DAMPER SERVOACTUATOR

CONCEPT 48-092 ACTIVE STANDBY WITH ELECTROHYDRAULIC VALVES 680J

This system is basically composed of four electro-hydraulic pressure control valves, a main stage servo valve, four hydraulic comparators, and two switching valves, Figure A-68. The main servo valve has piston areas with 2 to 1 ratio and is spring loaded to the center (neutral) position. During normal operation the four electro-hydraulic servo valves are operated and produce an output pressure which is a function of their driving signal. The comparators monitor the difference between the four pressures while the pressure from one of the valves controls the main stage servo valve position. In the event that a discrepancy occurs between the main stage servo valve control pressure and that of the comparator channels, the switching valve actuates and exchanges functions between the active channel and one of the standby channels. Three of the electro-hydraulic servo valves can be engaged in this manner while the fourth functions only as a monitor.

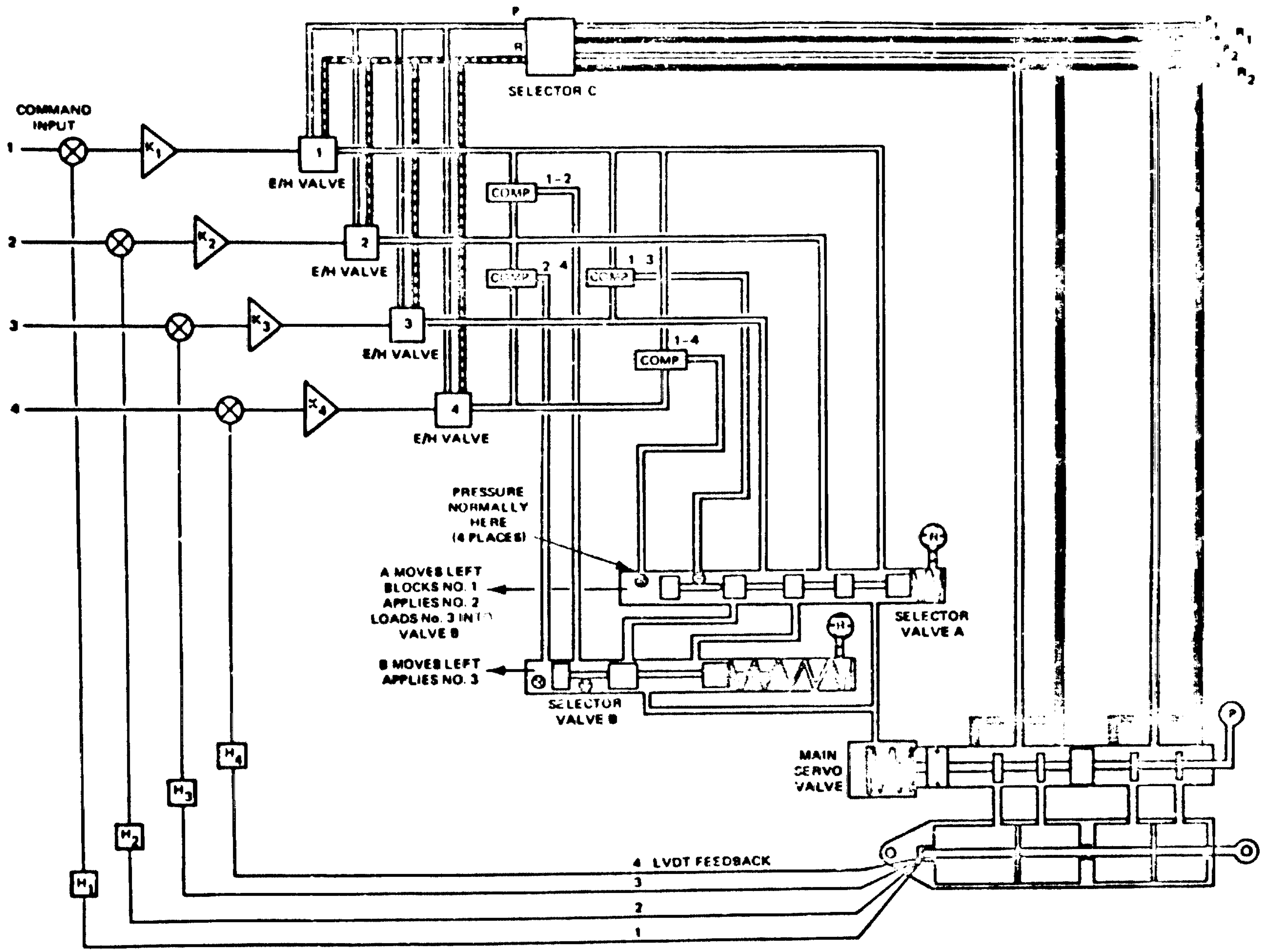

FIGURE A-68 ACTIVE STANDBY WITH ELECTROHYDRAULIC VALVES

CONCEPT 49-005 TRIPLE REDUNDANT SERVO SYSTEM

The Spool-Monitored (Electrical) Standby Redundant Actuator configuration, Figure A-69, uses three real channels (identical within tolerances) and a model channel which may be hydraulic or electronic. One real channel is active while the other two operate in standby. Actuator position feedback is electrical. The two-stage servovalves are coupled to the actuator through a four-position engage valve (or through 2 two-position engage valves). The engage valve transfers the system through its operational modes on commands from the electronic monitor via electrohydraulic solenoids. Position transducers on the servovalve spools and model channel (or equivalent) provide signals for comparison monitoring. Complete channel isolation is maintained. Failures are detected without requiring actuator motion from the command position.

For the first failure, the system switches from the active channel to a standby channel unless the failure is in a standby channel. In this case, the monitor prevents that channel from ever being engaged. For a second failure, the system switches to the remaining standby channel. In any operational mode, the engage valve bypasses the nonengaged actuators.

The configuration has the following advantages:

(1) No performance degradation due to failures since each channel can carry the load.

(2) Failure isolation maintained.

(3) Mechanization easily expanded to higher redundancy.

(4) Eliminated mechanical linkages.

(5) Actuator deviation is not required for failure detection.

The configuration has the following disadvantages:

(1) Depends on solenoid valve and engage valve reliability for transfer.

(2) Fast transfer times require high speed solenoids and comparators to minimize transients.

(3) Monitor may be sensitive to large power transients.

(4) Large size and weight because each actuator must be sized to carry the full load.

(5) Monitor sensitive to large load variations because of the electronic model.

(6) Spool position transducer reduces valve performance, increases cost, and lowers reliability.

Comments:

This configuration could be designed for active redundancy using synchronization to eliminate disadvantages 2 and 4 but at the expense of added complexity.

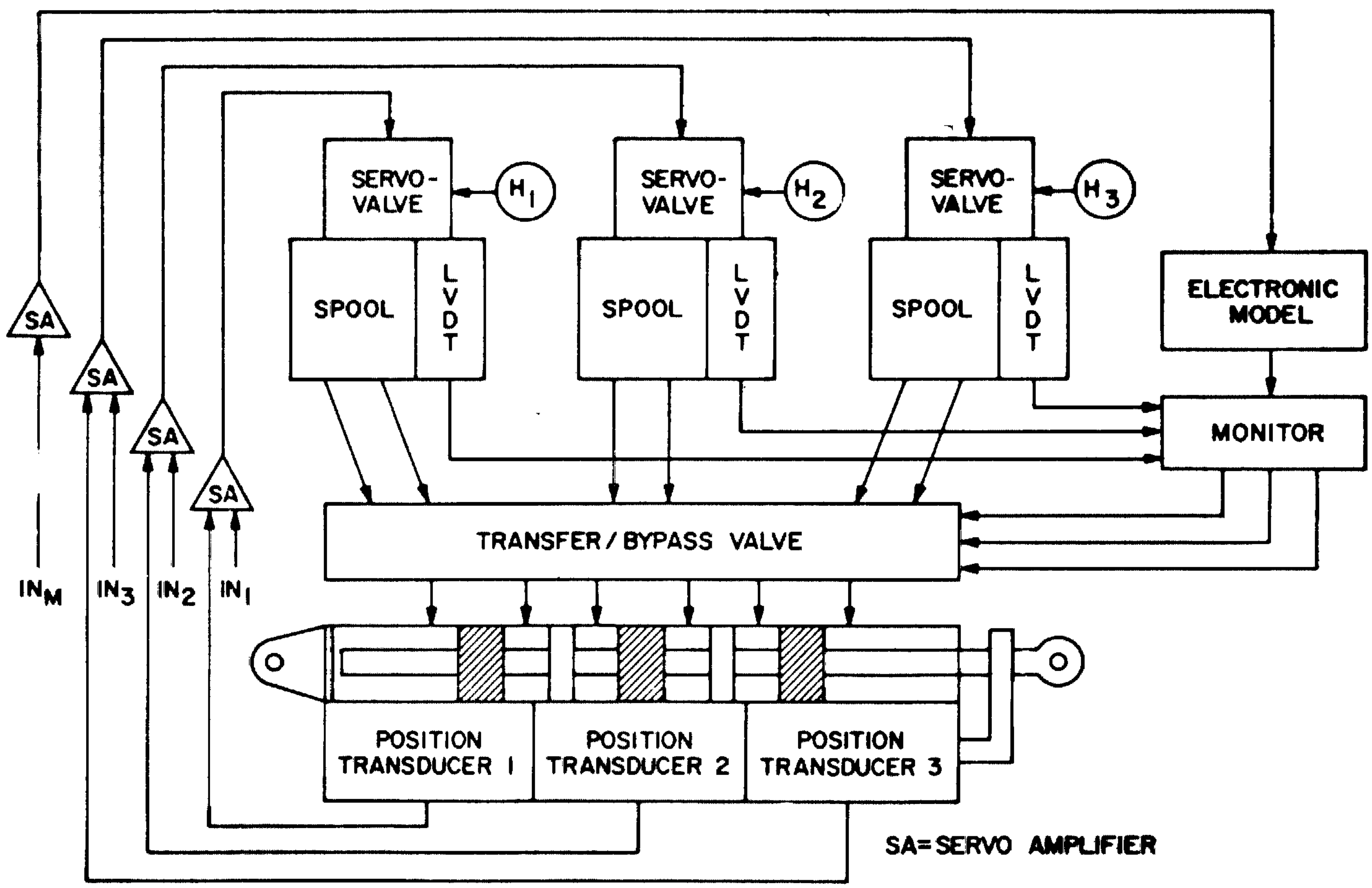

FIGURE A-69 SPOOL MONITORED (ELECTRICAL) STANDBY TRIPLE REDUNDANT ACTUATOR

CONCEPT 50-005 TRIPLE REDUNDANT SERVO SYSTEM

The Spool-Monitored (Hydraulic) Standby Redundant Actuator configuration, Figure A-70 is very similar to 49-005 Concept except for the monitoring mechanization which is all hydraulic. This configuration uses four real channels (identical within tolerances) with one acting as a model. Actuator position feedback is electrical. One channel is active while all others are in standby. The servovalves are coupled to an actuator through a four-position engage valve. The engage valve transfers the system through its operational modes on commands directly from the hydraulic monitor. The positions of the servovalve spools are measured and compared hydraulically. Complete channel isolation is maintained. Failures are detected without required actuator deviation from the commanded position.

For the first failure, the system switches from the active to a standby channel unless the failure is in a standby channel. In this case the monitor prevents that channel from ever being engaged. In any operational mode, the engage valve bypasses the nonengaged actuators.

The configuration has the following advantages:

(1) No performance degradation due to failures since each channel can carry the load.

(2) Failure isolation maintained.

(3) Monitor insensitive to load variations.

(4) Mechanization easily expanded to higher redundancy.

(5) Eliminates mechanical linkages.

(6) Actuator deviation is not required for failure detection.

(7) System transfer is very fast because electrohydraulic solenoids are not used.

The configuration has the following disadvantages:

(1) Depends on comparator and engage valve reliability for transfer.

(2) Hydraulic comparators are not fail-safe.

(3) Increased size and weight because each actuator must be sized to carry the full load.

(4) The close electrical tolerance required to match channels for small failure monitoring is costly.

(5) Hydraulic logic is susceptible to silting effects.

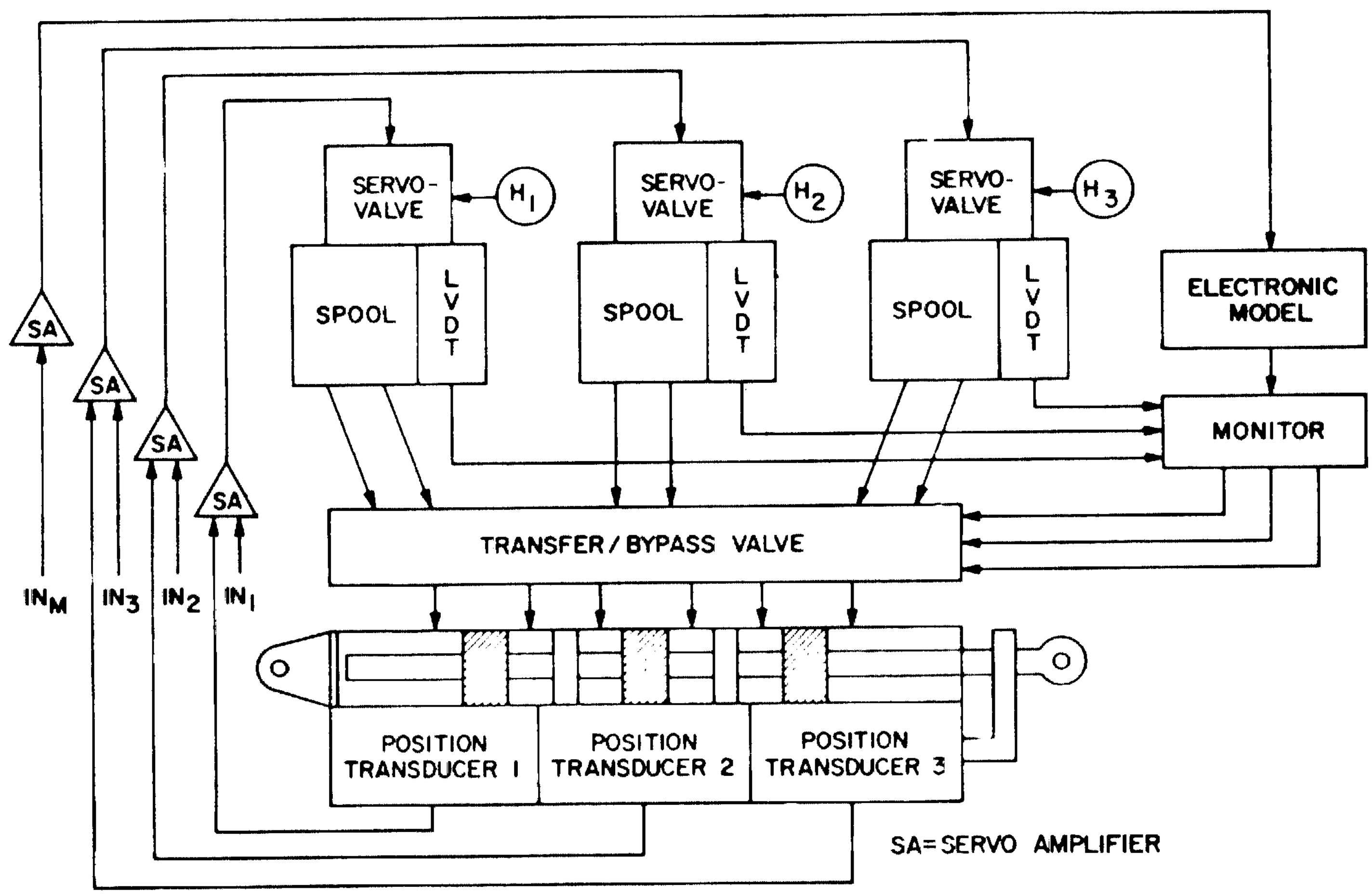

FIGURE A-70 SPOOL MONITORED (HYDRAULIC) TRIPLE REDUNDANT ACTUATOR

CONCEPT 51-109 C-5A INBOARD ELEVATOR SERVO

Functional Description. The Pitch Augmentation (P/A) provides pitch damping as shown in Figure A-71. A constant gain of 0.5 degrees inboard elevator per degree pitch attitude per second is used for manual flight throughout the envelope. When the Pitch Autopilot is engaged, the gain automatically increases to 2.0 degrees per degree per second. This high gain provides G-limiting needed to protect the aircraft structure in the event of a Pitch Autopilot hardover failure at high speed.

Analog Mechanization. The P/A subsystem is designed to be fail operational and achieves this objective by utilizing triple redundant input channels and dual-dual output channels. Figure A-72 is a hydraulic schematic of the mechanization.

Input Channels. The input channels consists of three identical sets of the following:

- Pitch rate gyros
- Body bending filters
- Gain select circuits
- Limiters

Comparators are used to detect malfunctions of the input channels by continuously comparing their relative output voltages through a preset voltage threshold.

The pitch rate gyro signals are processed through body bending filters to preclude the system from exciting or sustaining structural bending and are then sent to gain select circuits used to change the pitch rate gain from 0.5 to 2.0 degrees per degree per second when the autopilot is engaged. The signals are then limited to maximum SAS authority and provided to the output channels. The P/A receives two autopilot engage signals when the autopilot engage switch is depressed. The engage signals pull in relays and their contacts are used to ground the inputs to the gain switch drivers. The switch drivers in turn open the transistor switch in the gain select amplifier to remove a resistor from the circuit. With the resistor out of the circuit the gain increases from 0.5 to 2.0 degrees per degree per second. In order to obtain a third autopilot engage command for pitch rate, the two autopilot engage signals are ANDed and if both are present, the third switch driver input is grounded. Once grounded, the switch drive changes the gain select amplifier gain in the same manner as before. An extra set of contacts on relays are used in conjunction with the system engage relays to provide validities to the autopilot. In the event the P/A disengages or does not change pitch rate gain, the validities to the autopilot are interrupted and the autopilot thereby disengages.

Output Channels. Four median selectors and four servo amplifier/feedback loops are utilized in the output channels.

Two of the four output channels are used for Channel A and two for Channel B. Under normal operation only the Channel A series switches are closed and the median selectors' signals, through the Channel A servo amplifiers, are used to drive the Channel A electrohydraulic valves on each servo. SAS actuator position feedback is provided by a linear variable differential transformer on each actuator. In the event a failure occurs in one of the Channel A servo loops, the servo comparator will trip and the monitoring logic will open the Channel A series switches and disengage the Channel A servo loops. If the channel B monitoring logic is valid (no failure), the channel B series switches are closed and the channel B median selectors and servo amplifiers are used to drive reserve electrohydraulic valves on each servo. Should a subsequent failure occur in one of the channel B servo loops, another comparator will trip and the monitoring logic will disengage the entire system and send a P/A inoperative signal to the annunciator panel and the Flight Augmentation Control Panel.

Monitoring Logic. The P/A computer contains continuous monitoring which is capable of detecting the occurrence of faults which could degrade the performance of the subsystem. The following parameters are continuously monitored:

a. Input Channels
b. Servo Loops
c. Hydraulic Pressure
d. Electrical Power
e. LVDT Excitations

The logic mechanization consists of two channels and is designed to insure that a single logic failure cannot prevent disengagement in the event of multiple system failures. This objective is achieved by using dual logic channels for channel A servo loop engagement and dual logic channels for channel B engagement. Both halves of a logic channel must be valid to maintain the respective servo loop channel engaged.

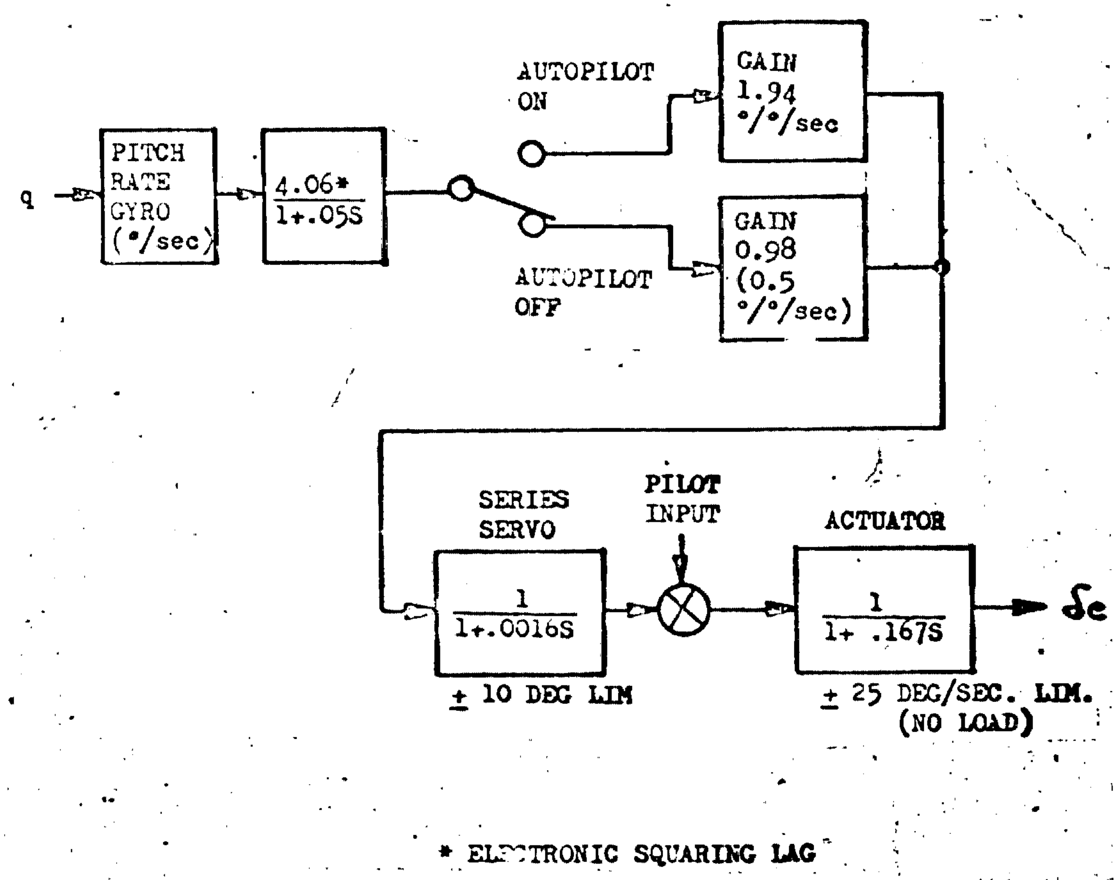

FIGURE A-71 PITCH AUGMENTATION SUBSYSTEM BLOCK DIAGRAM

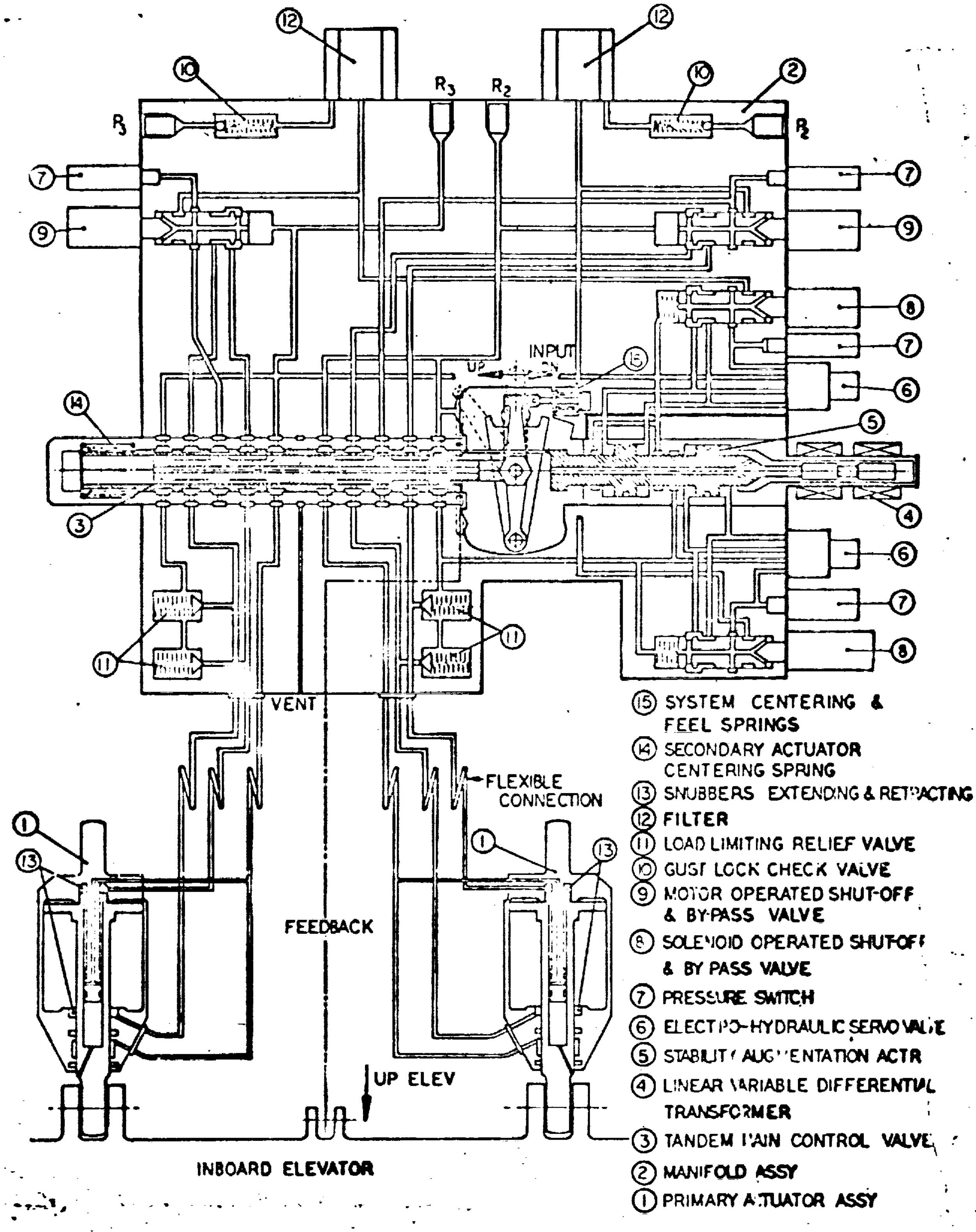

FIGURE A-72 HYDRAULIC SCHEMATIC INBOARD ELEVATOR SERVO

CONCEPT 52-110 DIRECT DRIVE ANALOG VALVE

A model 700 Electrohydraulic Servo Valve manufactured by Pacific Controls Incorporated is shown in Figure A-73.

Design Features:

- Spool directly driven by patented high force output linear motor.
- No filters, nozzles, orifices, or jet pipe to plug or clog.
- Valve characteristics not affected by inlet pressure. (Response, hysteresis, threshold and null)
- Stainless steel spool and sleeve.
- Encapsulated coils, Teflon insulated internal wiring.
- Patented force motor of symmetrical mechanical design for excellent null stability.
- Simple design, yielding LOW COST and reliability.
- Null or other characteristics not affected by adjacent ferrous materials or magnetic devices.

Performance Characteristics:

Operating Pressure Range	0 to 2000 Psi
Return Proof Pressure	2500 Psi
Rated Coil Current	1000 Ma Diff. Current ±10%
Coil Resistance	7 Ohms ±10%
Threshold (From 0 to 2000 Psi. inlet press.)	.5% Max. of rated current
Hysteresis (From 0 to 2000 Psi. inlet press.)	5% of rated current
Force motor stall force	110 lbs. (±55 Lbs.)
Frequency Response (From 0 to 2000 Psi. inlet P)	90° at 230 Hz
Null Stability (From 0 to 2000 Psi. inlet Press.)	Within ±3.5% of rated current from -10°F to 220°F
Flow Rate capability	.5 GPM to 12 GPM at 2000 Psi.
Flow linearity	15%
Null leakage	.05 to .1 GPM less than nozzle flapper or jet pipe two stage valves at 2000 Psi.

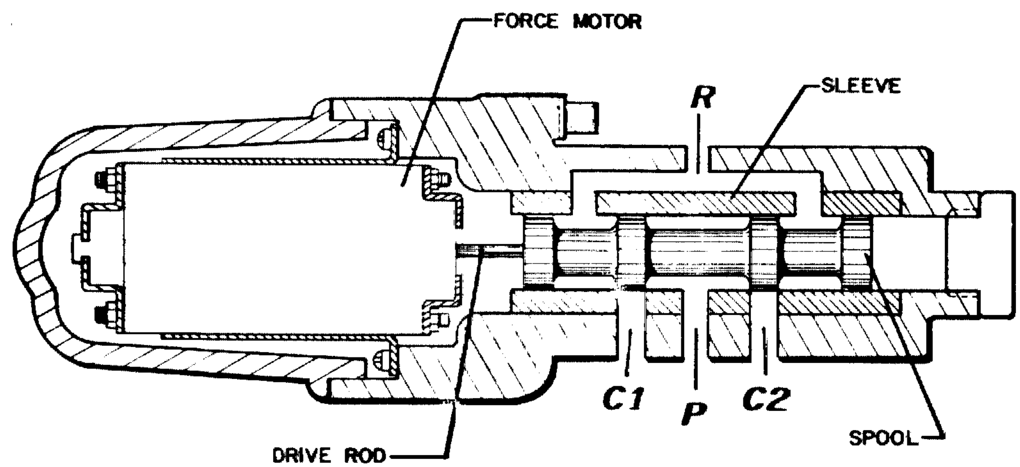

FIGURE A-73 ELECTROHYDRAULIC SERVOVALVE MODEL 700

CONCEPT 53-023 DYNAVECTOR ROTARY ACTUATOR

The Bendix Dynavector rotary actuator technology was demonstrated by the fabrication and evaluation testing of several prototype actuator test articles with the final evaluation being conducted on a 100,000 in-lb capacity unit designed to meet an assumed set of flight test performance requirements.

The fluid power Dynavector motor is an integral high-speed motor and high-ratio transmission without high-velocity mechanical elements. (Figure A-74). The major components of the Dynavector motor assembly consist of a series of displacement chambers, a unique integral epicyclic transmission, and commutation porting. The transmission and motor use elements common to both, resulting in a much simpler and more reliable design.

Two features make this actuator small and light: (1) its transmission reduces the high-speed input to a low-speed output in one step, using only two moving gears; and (2) it has no physical motor--only a rotating force vector; hence, the name "DYNAVECTOR." As the force vector rotates at high speed, it causes the input member of the transmission to orbit. This orbiting member is geared directly to the rotating output shaft. Because it orbits instead of rotates, gear-tooth contact velocities are small and a large speed reduction is accomplished without complex gearing.

The power element is a positive displacement, very low inertia, non-rotating vane motor. Its output is a radial force vector which rotates at high speed and in either direction of rotation. The displacement chambers formed by the vanes and the housing expand and collapse at the same speed as the force vector, but do not rotate. The motor is self-commutating but does not contain a rotating porting plate or spindle. The absence of high-velocity members in the motor significantly reduces the inertia, resulting in high acceleration capability.

The integration of the power element and epicyclic transmission into an integral actuator design results in an ideal servoactuator with a high torque-to-inertia ratio and high efficiencies at rated loads.

Specific advantages of this new actuator concept include:

Improved Performance

The motor's new operating principle reduces the actuator inertia (as seen at the output shaft) more than 100 times. As a result, dynamic frequency response improvements of more than one decade can be obtained.

Decreased Weight and Size

The complete actuator, including transmission and motor, weights only as much as the assembly of a conventional transmission that is equally rated. Design flexibility allows almost any envelope requirement to be met.

CONCEPT 53-023 (Continued)

Lower Cost

Basic simplicity results in significant manufacturing cost reductions, compared with conventional units. The operating principle demands less critical tolerances, resulting in further cost reduction.

Higher Reliability

Extremely low relative velocities between all moving members reduces wear to a minimum. Reduction in number of parts enhances reliability over more complex conventional systems. All members are rigid, to eliminate fatigue limitations.

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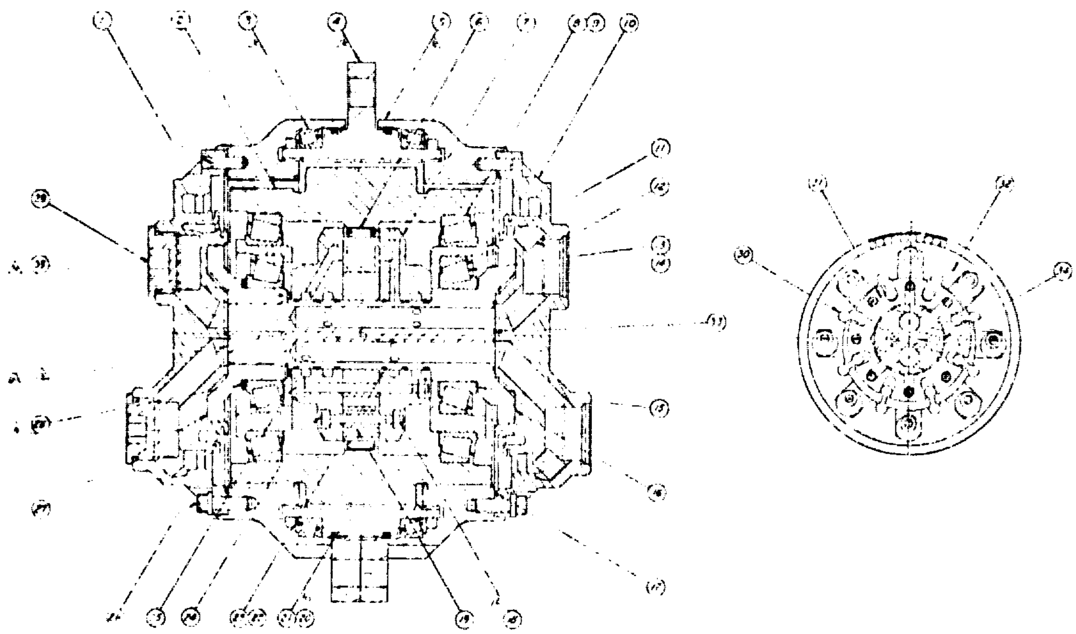

ITEM NO	REQD	PART NO	DESCRIPTION
1	2	Stock	Dowel
2	1	2175115	Ring Gear
3	2	Stock	Roller Bearing
4	1	2158408	Output Mtg Gear
5	2	2165464	Ground Mtg Gear
6	1	Stock	Needle Bearing
7	2	2172181	Manifold
8	2	Stock	Roller Bearing Cone
9	2	Stock	Roller Bearing Cup
10	2	2169595	End Cap
11	2	2175114	Eccentric Ring
12	2	Stock	Plug
13	2	Stock	Roller Bearing Cone
14	2	Stock	Roller Bearing Cup
15	1	2175128-1	Shim
16	1	2172432	Motor Shaft
17	2	Stock	O-Ring
18	16	Stock	Hex Nut
19	8	2172428	Motor Spacer
20	2	Stock	Quad Ring
21	2	Stock	Spiral Backup Ring
22	4	Stock	O-Ring
23	6	Stock	Backup Ring
24	8	Stock	Hex Socket Head Screw
25	8	Stock	Hex Socket Head Screw
26	20	Stock	Tension Bolt
27	1	2171865	Center Plug
28	1	2175128-2	Shim
29	2	Stock	Plug
30	8	2172429	Motor Vane
31	1	2172431	Motor Spacer
32	1	2172433	Reaction Ring
33	2	Stock	O-Ring
34	1	2172434	Motor Key
35	1	2175128-3	Shim
36	Ref	2173210	Output and Grd Gear

FIGURE A-74 HH-267-U3 DYNAVECTOR ACTUATOR

CONCEPT 54-024 ROTARY ACTUATOR FOR SPACE MISSIONS

This actuator is a unique integrated motor and epicyclic gear reducer. It differs from previous epicyclic reducers in that the motor is functionally integrated with the transmission. (Figure A-75 shows the breadboard actuator).

Figure A-76 illustrates the operating principle of the device. The armature of the motor and the ring gear of the transmission are combined in a single element which is driven by the attractive force of a rotating magnetic vector in the stator. For each vector rotation, the ring gear completes an eccentric cycle. Because the number of teeth on the ring gear is different from that on the fixed ground gear, the ring gear rotates a fraction of a revolution. The ring gear simultaneously engages the output gear, which in one eccentric cycle is rotated a fraction of a revolution in the opposite direction. Since the pitch diameters of the two meshes are not the same, there is a resultant differential motion of the output gear.

The advantages of the design are--

Only two bearings are required, and these are on the low speed output shaft.

The fixed ground gear and the output are bridged by a single stiff member (the ring gear), resulting in extremely low windup.

The gear meshes are floating, which allows them to self-center. Since allowance for fixed shaft center distance tolerances is not required, an inherent minimum backlash configuration is achieved.

A high reduction ratio per number of meshes is obtained; for example, 818 to 1 with only 2 meshes.

The new actuator can be driven digitally for ultraprecise numerical control systems, or it may be driven as a brushless dc motor with ideal characteristics for use in linear servomechanisms. This actuator represents a significant advancement in the art of gear reduction.

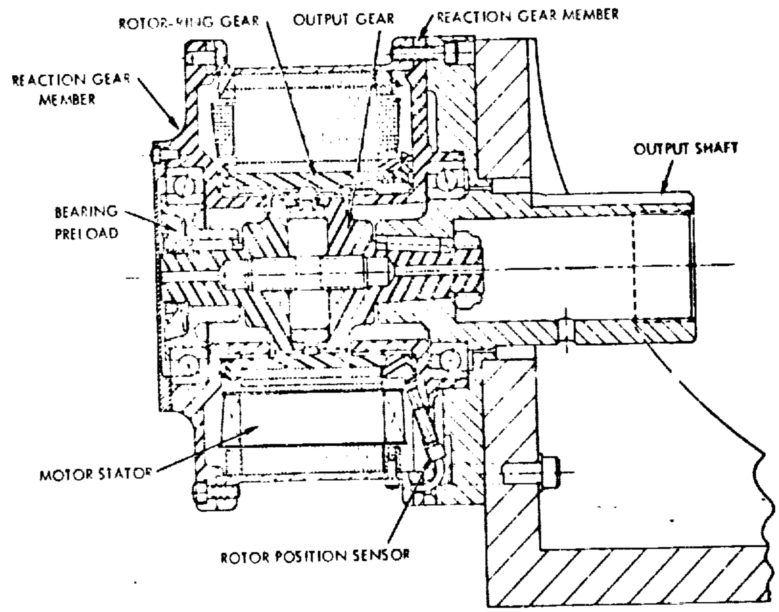

FIGURE A-75 BREADBOARD ACTUATOR

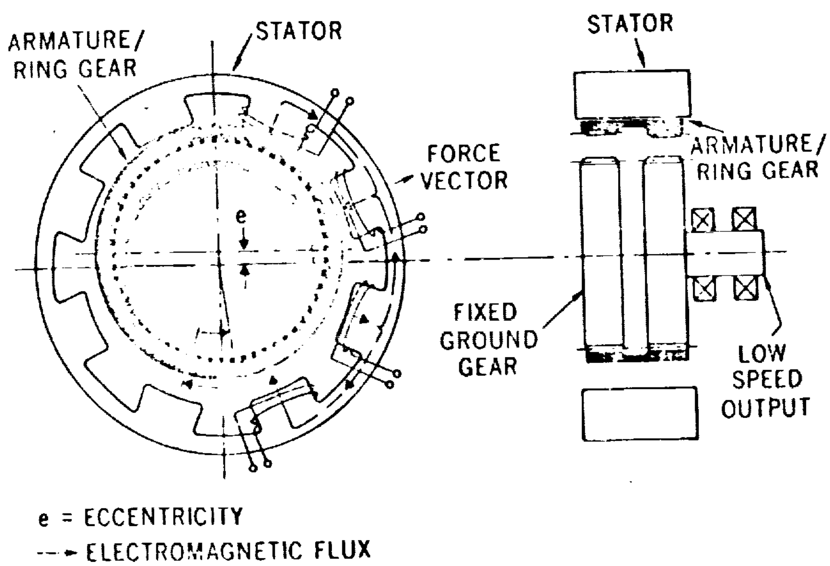

FIGURE A-76 OPERATING PRINCIPLE OF ROTARY ACTUATOR

CONCEPT 55-149 EQUIPMENT ON B-58, F111, F-14, 747 & DC-10

The Bendix Electrodynamics Division's actual aircraft flight control experience began with its activity in the design, development, and production of the B-58 primary flight control system (Figure A-77). This equipment has been qualified for operation in a 350°F environment and has a proven reliability index of less than 10 failures per million flight hours.

Electrodynamics also performed the design, development, and production of the F-111 primary flight control system. This equipment has been qualified for operation in the F-111's Type II hydraulic system.

Similar equipment is also being produced for the F-14 primary flight controls systems (Figure A-78). This equipment is currently undergoing qualification testing.

Electrodynamics also is producing and has qualified the inboard and outboard elevon actuators for the 747 jumbo jet transport (Figure A-79).

The flight control equipment currently being produced and qualified for the McDonnell Douglas DC-10 is shown in Figure A-80.

Electrodynamics designed the SST rudder servo actuators prior to this program's cancellation. These units consisted of three individual servo actuators, synchronized to drive a single control surface panel. The rudder contained three such control panels. The servo actuators were unique in that they were triple redundant with both structural and hydraulic isolation. The actuators were titanium fixed body units that use an internal flex rod. Design temperatures are -20° to 450°F. The units had a design life of 32,000,000 cycles and 50,000 hours.

Electrodynamics' most recent efforts include the design, development, and manufacture of the YF-16 (lightweight fighter) horizontal stabilizer, rudder and flaperon servo actuators for the prototype fly-off program.

The Division has also designed, developed and is currently building the YC-15 (AMST) aileron and rudder servo actuators for the prototype fly-off program.

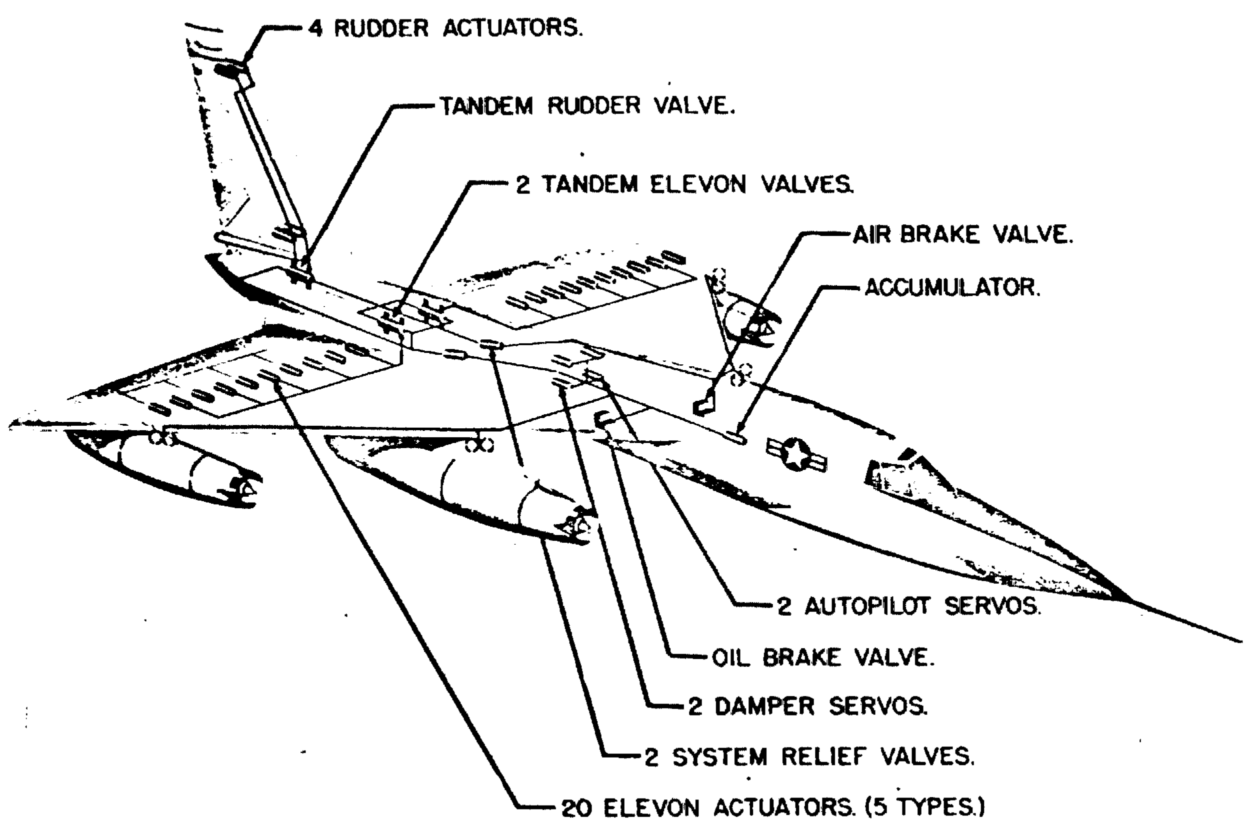

FIGURE A-77 BENDIX EQUIPMENT ON B-58

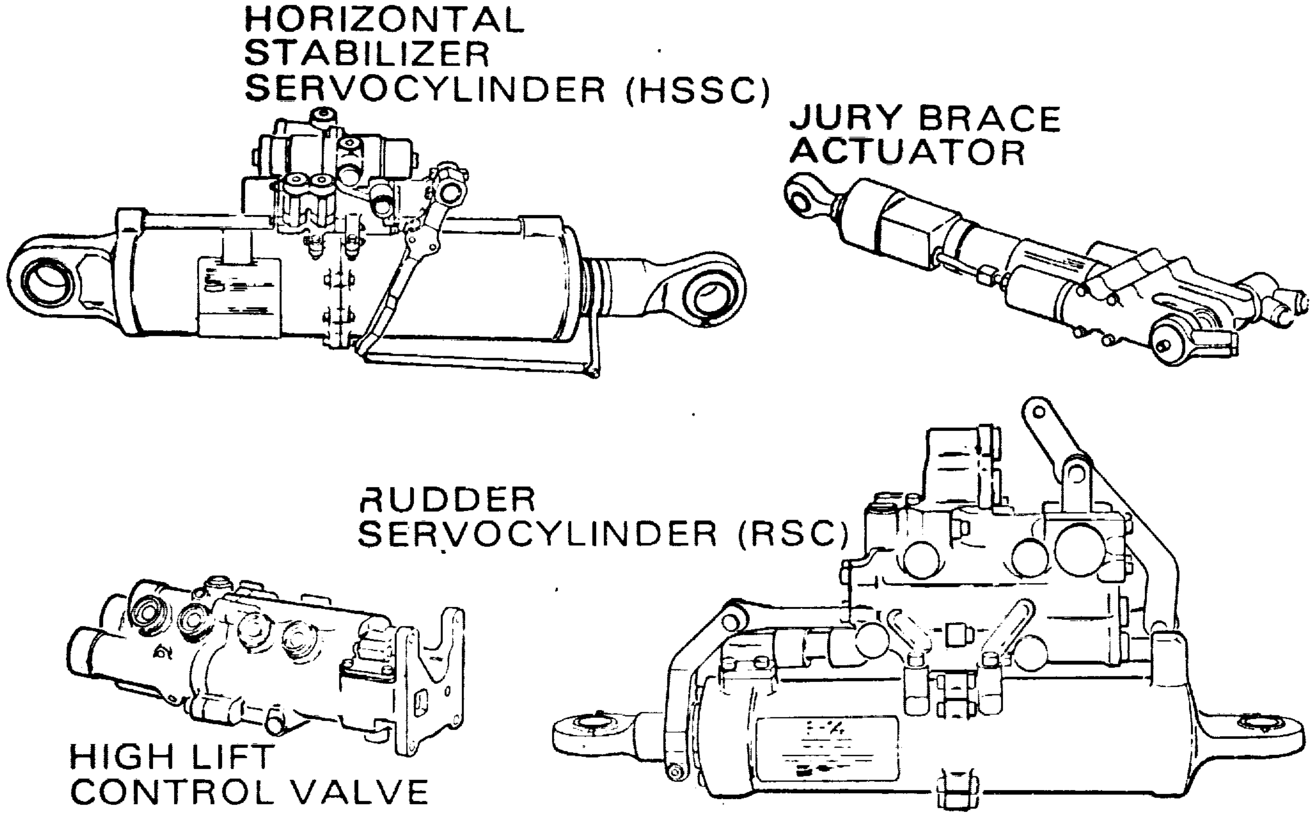

FIGURE A-78 BENDIX EQUIPMENT ON F-14

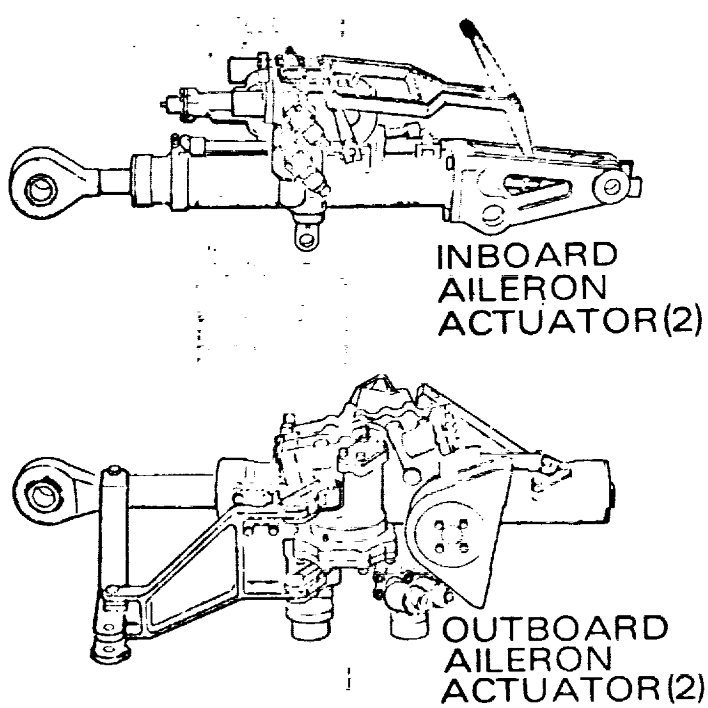

FIGURE A-79 BENDIX EQUIPMENT ON 747

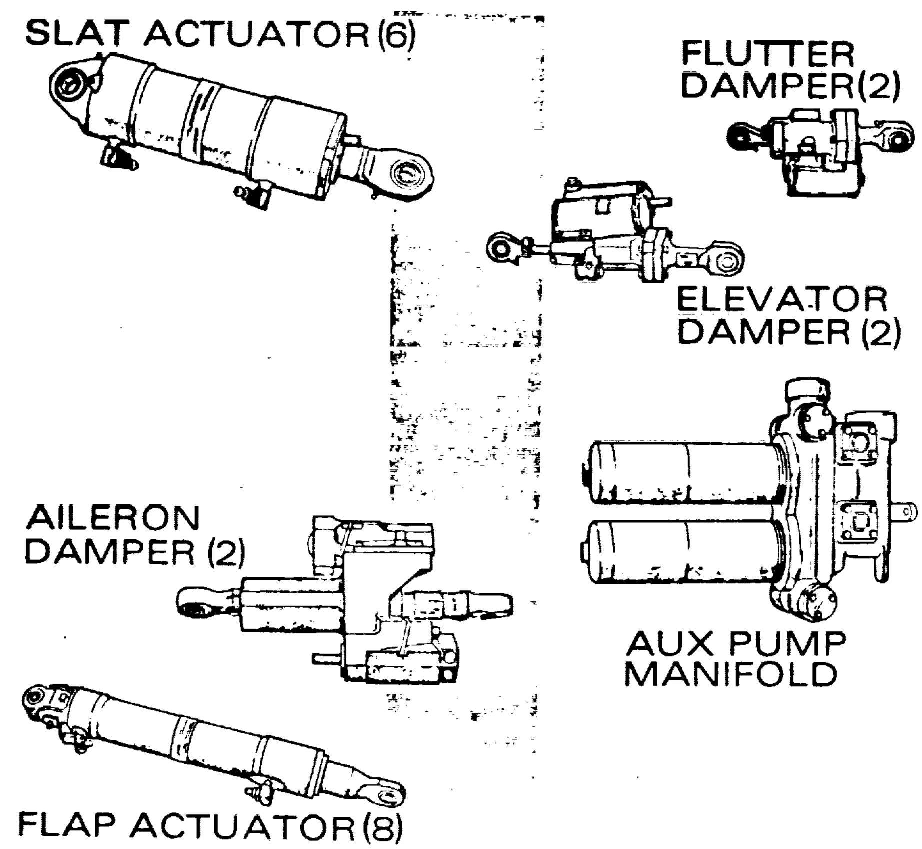

FIGURE A-80 BENDIX EQUIPMENT ON DC-10

CONCEPT 56 -123 MECHANICAL FEEDBACK ACTUATORS (MFB)

The basic improvement offered by MFB actuators is a significant increase in system reliability achieved through elimination of the electrical position transducer (potentiometer, LVDT, or other) together with its associated cabling and power supply. See Figure A-81. Potentiometers have long been a serious limitation on the life and environmental capabilities of servoactuators. Substitution of an LVDT imposes additional electrical complexity and still leaves the feedback dependent upon an electrical power supply. Elimination of the electrical transducer altogether is highly desirable.

Even more significant is the drastic loss of control resulting from electrical failure in a conventional electrical feedback actuator. If the potentiometer or LVDT opens, or if one or more wires in the connecting cable are severed, or if part or all of the electrical supply is lost, the actuator will drive hardover, usually at high velocity. Also with an electrical feedback actuator, the servoamplifier is located within the servoloop and loss of the amplifier or its connecting cables causes an open loop failure.

The MFB actuator, on the other hand, will "fail-neutral" with loss of electrical control.

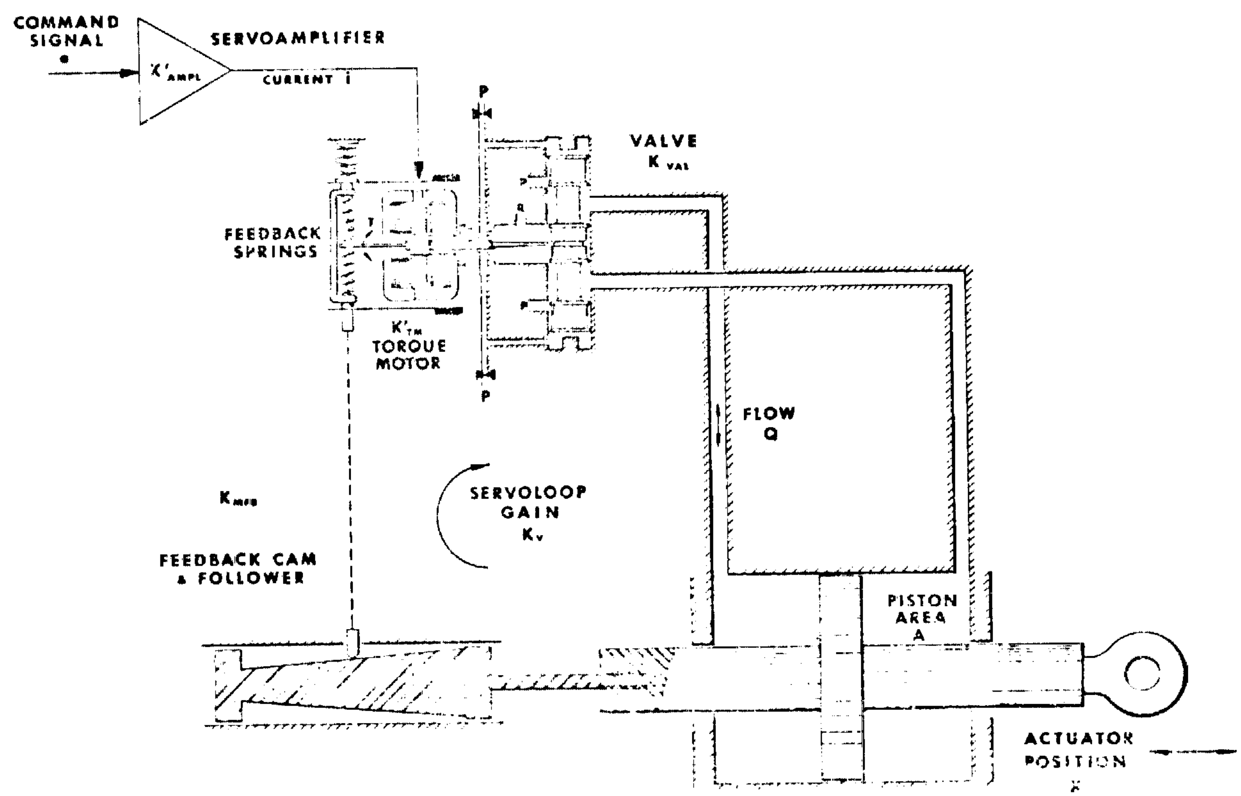

FIGURE A-81 MECHANICAL FEEDBACK ACTUATOR WITH TWO-STAGE SERVOVALVE

CONCEPT 57-151 FLUIDIC SERVOACTUATOR AND ELECTROMAGNETIC INPUT TRANSDUCER

An investigation effort was concerned with the feasibility demonstration of a new and unique servovalve and servoactuator concept, which used fluidic elements and circuits in their design. In conjunction with the fluidic servovalve, a fluidic feedback transducer is being developed, which will eliminate energy conversion components that involve multiple interface signal conversion from hydraulic to mechanical to electrical. The fluidic valve and transducer are shown schematically in Figure A-82.

A model of the fluidic valve was fabricated and the feasibility of this concept successfully demonstrated with liquid metal NaK-77. This development offers a servoactuating subsystem that has only two moving parts: the actuator rod and the second stage valve spool. All other functions are provided hydraulically by fluidic devices.

Prior to building a liquid metal integrated package, a number of intermediate studies were initially required; hence fluidic control with MIL5606B oil was investigated. Under severe separately sponsored General Electric studies, these fluidic concepts were appropriately modified for use with conventional hydraulic oils by substitution of the MHD effect first stage unit with an EM pump-bellows-flapper nozzle element arrangement, Figure A-83. The breadboard model of the oil version of this valve was also fabricated and evaluated. This valve was mounted to a modified F-4 elevator actuator. The closed-loop performance characteristics obtained from this breadboard unit not only offers new possibilities for the NaK system, but also shows great promise for future flight control systems operating in the medium temperature range (300-600 F).

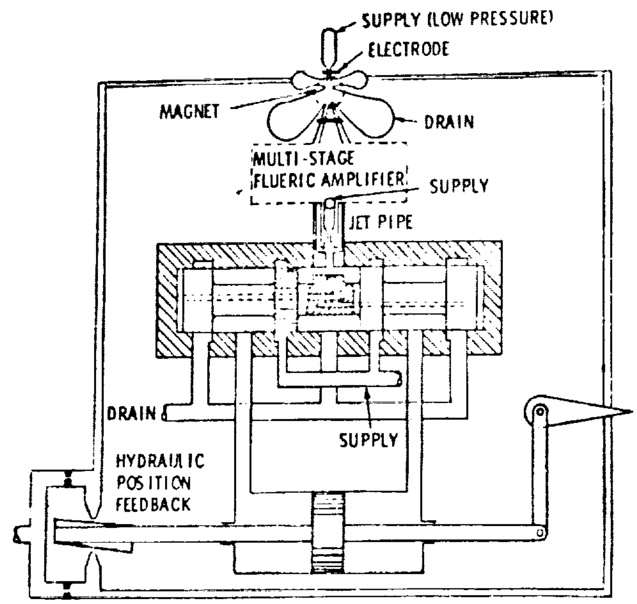

FIGURE A-82 FLUIDIC SERVOACTUATOR SCHEMATIC

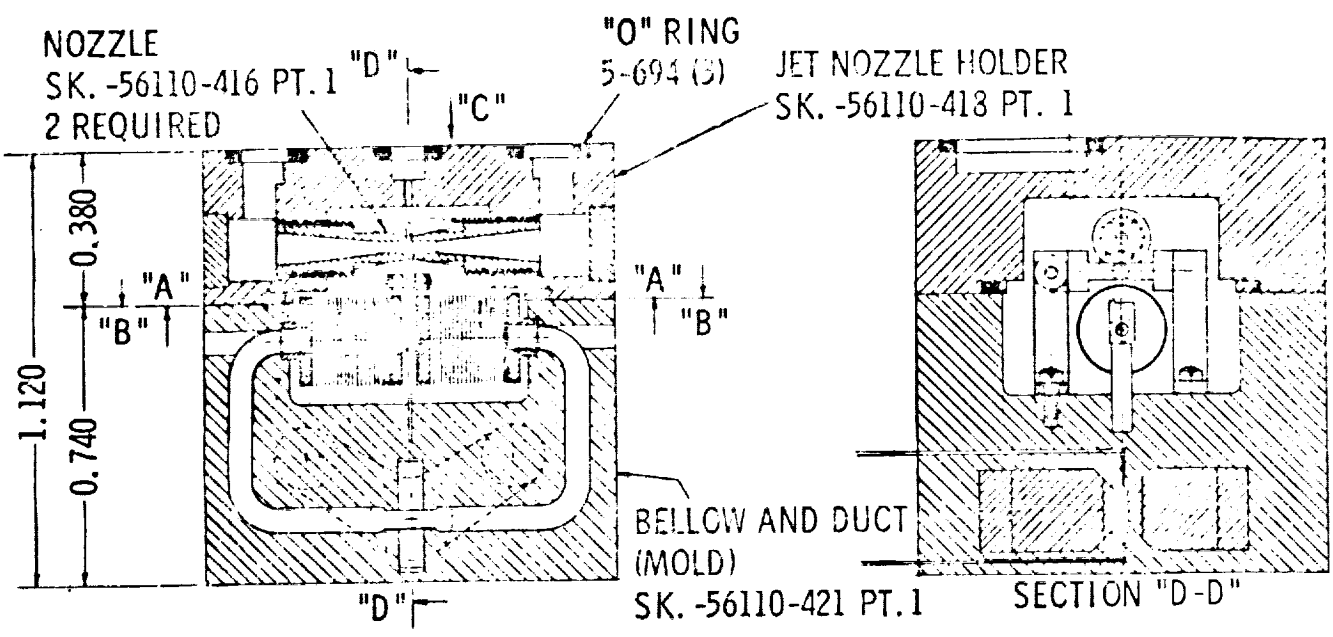

FIGURE A-83 ELECTROMAGNETIC INPUT TRANSDUCER

CONCEPT 58-152 MISSILE FLIGHT CONTROL ACTUATOR

The actuation system positions each of four fins independently in response to command signals from the missile autopilot and provides electrical fin position feedback signals to the autopilot. The prime source of power is a battery. The actuator is designed to bolt onto an internal mounting pad on the missile airframe. Each actuator is made up of three subsections: the motor assembly, the servo assembly, and the fin housing assembly.

The motor assembly consists of a D.C. drive motor, speed switch, flywheel, electromagnetic interference suppression filter and electrical connector. The servo assembly consists of drive gearing, spring clutch servo, and output screw jack. The fin housing assembly contains a rack and sector gear, fin shaft socket with integral fin shaft lock, feedback transducer and electrical connector.

A schematic of the servo actuator is shown in Figure A-84. The D.C. drive motor, through appropriate reduction gearing, continuously drives two spring clutch input drums in opposite directions. When a command signal is applied to one clutch energizing magnet coil, rotation of the control clutch output disc causes the clutch spring to engage the input drum. The spring, operating on a capstan principle, acts as a torque amplifier; therefore, the torque necessary to energize the spring is only a fraction of that which can be transmitted. The output end of each clutch spring is attached to the rotating nut of the screw jack.

An integral bearing supports the screw jack thrust load. The translating screw jack shaft drives a rack and sector gear which rotates the missile fin shaft. By using two spring elements, the servo provides bi-directional output. When the command is removed, the springs act as brakes locking the output to fixed structure. Limits on the screw shaft mechanically declutch the clutch springs at each end of the stroke by mechanically operating the control clutch. A mechanical interlock interconnects the energizing end of each spring to prevent simultaneous engagement of the clutches. The position transducer is coupled to the screw shaft for generation of the feedback signal. The transducer is connected to the translating screw shaft rather than the rotating fin shaft in order to provide a more stable loop immediately around the servo. If the transducer is connected at the fin the added mass dynamics may have a destabilizing effect on the servo loop. With the transducer connected at the translating shaft, the servo loop is stiffer while the effect of the mass dynamics of the fin shaft has a negligible effect on the outer loop. In either case the effects are small and the convenience of mounting the transducer in the servo housing is significant.

As noted above, the motor assembly contains an integral flywheel. By design adjustment of the motor total shaft inertia (armature plus flywheel) and the motor speed-torque characteristic, a considerably small motor may be utilized than if no flywheel effect is used. This results in a significant reduction in motor current and smaller battery. Overall system weight is thereby reduced.

CONCEPT 58-152 (continued)

The motor also contains a centrifugally activated switch which closes when motor speed reaches approximately 80% of no load speed. Closure of these contacts is used in the missile launching sequence to signal that the actuators are up to speed and ready to operate.

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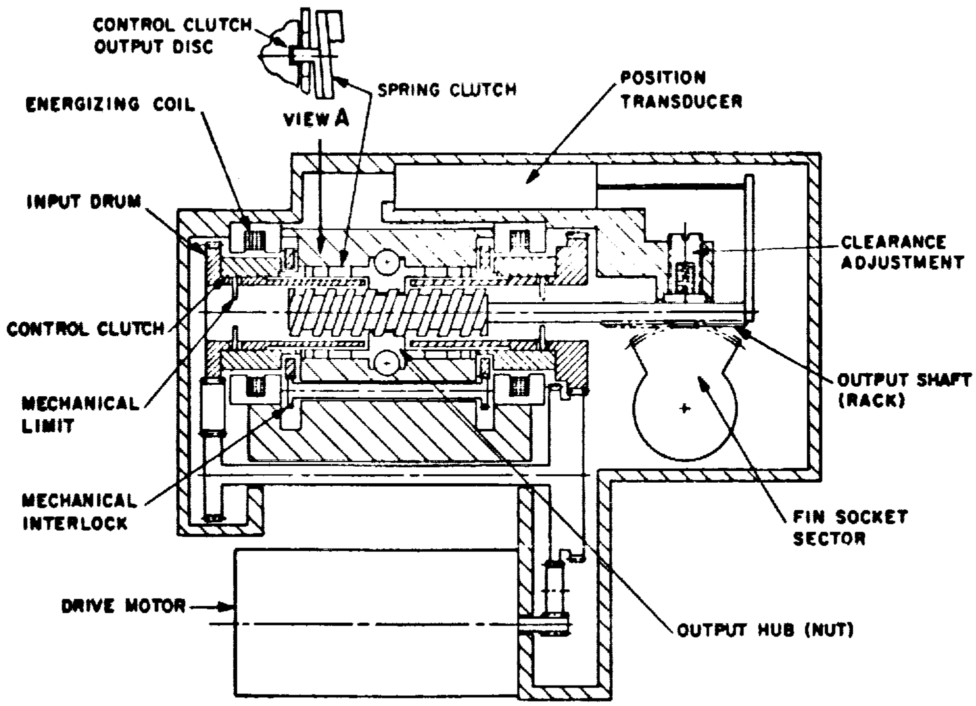


FIGURE A-84 SCHEMATIC-SERVO ACTUATOR

CONCEPT 59-129 INCREMENTAL ENCODERS

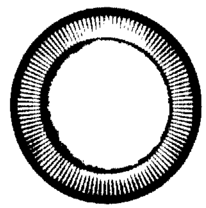
Gurley Incremental Encoders are photoelectric devices which convert shaft rotation into electrical signals. The Encoder consists of a light-tight housing which contains a glass disc having precisely-spaced clear and opaque segments on its periphery; source of illumination; slit plate; and photo sensors. Rotation of the disc produces quadrature pulses on each of two lines which may be used for pulse multiplication, direction-sensing, or both, by the use of external logic. An index pulse is an available option at additional cost. Sense-of-direction is achieved by the use of two outputs spaced 90° out-of-phase so that the direction of rotation can be determined by external logic circuitry.

The Teledyne Gurley Model 8706 incremental photoelectric linear encoder is designed for use wherever a linear change in position must be accurately determined electronically. This encoder features compact size and a wide variety of options in a standard reading head. Options include bidirectional sensing and one or two zero reference pulses, if required. Either photocell or photo transistor sensing can be provided.

Sense of direction is obtained by producing two signals in quadrature and is accomplished by inserting a phasing system in the optical portion of the system. This bi-directional capability and up to two separate zero reference points can be packaged within a one inch cube. The light source is collimated so that accurate readings may be made even though the gap between the head and scale varies from .003" to .010". This allows more latitude in set-up and lessens the risk of damaging the scale. The scale itself is chrome-clad for maximum ruggedness.

The Model 8706 encoder is available with pulse counts of up to 1000 pulses per inch. Accuracy of ± .0002" is standard.

The Teledyne Gurley Linear Encoder eliminates the need for high accuracy lead screws, racks and pinions, or similar devices normally used to translate linear motion into rotary motion. The linear encoder directly measures changes in linear position, which coupled with relatively wide gap tolerance, allows the simplest, most economical digital interface for every type of linear motion. Figure A-85 shows the typical encoder mechanization and standard disc configurations.

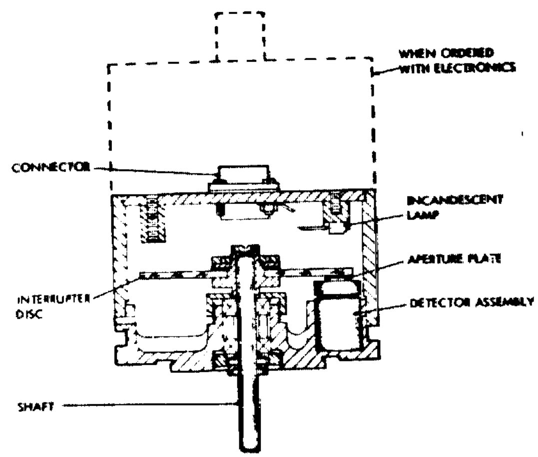


TYPICAL DISC PATTERN

STANDARD DISCS — PULSES PER REVOLUTION

1	500	1800	3600	100 w/index
10	600	2000	4000	360 w/index
36	792	2048	4096	1000 w/index
48	1000	2500	4800	2048 w/index
50	1024	3072	5000	3600 w/index
100	1100	3216		4000 w/index
120				4096 w/index
360				5000 w/index

ACCURACY = 3 MINUTES



TYPICAL ENCODER MECHANIZATION

FIGURE A-85 ENCODER MECHANIZATION

CONCEPT 60-130 TRIGAC I AND TRIGAC III TRANSDUCERS

The Singer Companies single channel TRIGAC III converter series constitute a new method of performing the synchro/digital conversion task with improved cost, size and accuracy features.

In contrast with current solid-state converters, these converters are ac rather than dc systems. There are no significant errors due to dc amplifier drift, switch offsets, or capacitor instability. Further, they utilize multiplication of the error signal by ac references, thereby eliminating errors due to harmonic generation in the input resolver. In addition to improvements of accuracy and lower cost, they can provide instant access to a number of synchro channels with staleness of the digital data virtually eliminated.

The problem of staleness of data is inherent in an electromechanical converter, i.e. there is a lag between digital output and shaft input because of the finite slew rate and response time of the follow-up servo. Most solid-state converters require at least one cycle of carrier frequency (nominally 2.5 milliseconds) for converting each input synchro. A typical coordinate conversion requires at least three and probably as many as five input angles. This means that some of the input data are at least ten milliseconds old. With the TRIGAC III converter, the input information is updated many times per cycle of carrier and the staleness is limited only by information bandwidth of the synchro carrier. In addition, all input data are avilable in a semiconductor memory and may be read out to the computer at any time. This capability means that this converter need not be operated synchronously or from the same clock as the digital computer using a multiplexed D/A converter.

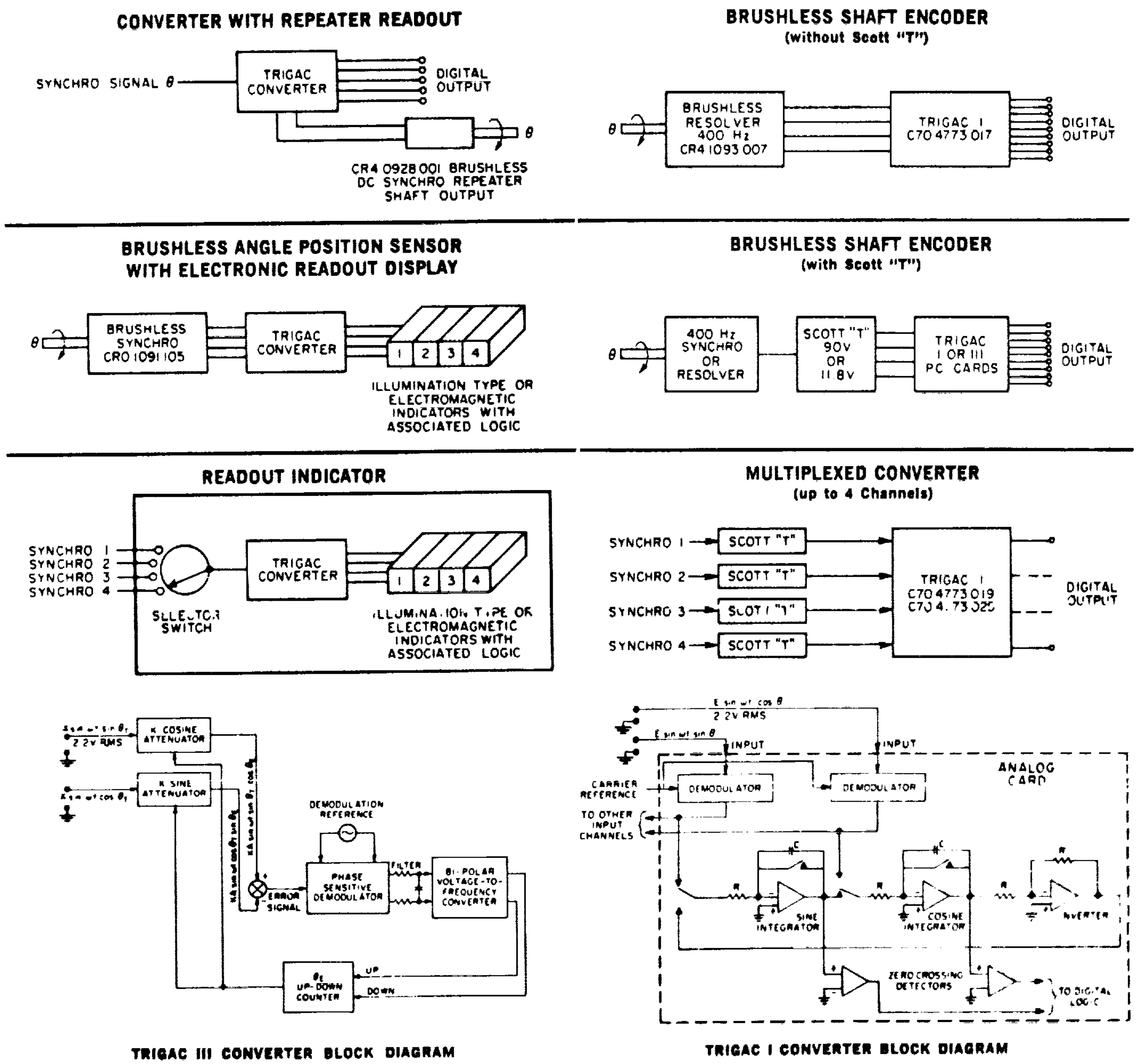
The single channel TRIGAC I converter is another versatile technique employing demodulators, integrators, inverters, and zero-crossing detectors. Like the TRIGAC III converter, it also features card construction to assure ease of maintenance and repair.

AC resolver signals, or synchro signals interfaced with a Scott-T, are demodulated to charge up two dc integrators in proportion to the sine and cosine of the input shaft angles. Integrators are subsequently connected in series with a unity-gain inverting amplifier in a closed loop, forming a two-phase oscillator. Oscillator period (typically designed to be 1/100 or 1/200 second) is determined by precision resistors and capacitors of the operational integrators, and is independent of carrier excitation frequency. A stable clock counts from the beginning of the oscillation until zero-crossing of either the sine or cosine voltages to provide the least significant bits.

Resolver waveforms are integrated, during the "initial condition" mode. "Oscillate" mode lasts for exactly ¼ oscillator period. Zero-crossing of either integrator during this interval is certain, and pulses accumulated until zero-crossing yields the input complement. Capacitors are then reset (shorted) until the next "initial condition" mode, as determined by carrier reference positive slope zero-crossing. A change in carrier frequency or amplitude changes only integrated signal amplitudes at the beginning of the "oscillate" mode, but not their ratio or total pulse accumulation.

CONCEPT 60-130 (continued)

Figure A-86 shows various system configuration and the respective block diagrams for these converters.



Accuracy of TRIGAC I is 12 minutes (±6 minutes), and of TRIGAC III is 2 minutes, 38.2 seconds (±1 minute, 19 seconds).

Resolvers are available with ±3 minutes accuracy.

FIGURE A-86 TRIGAC I AND TRIGAC III BLOCK DIAGRAMS

CONCEPT 61-C

TRANSDUCER

The G. L. Collins Linear Motion Transducer (LMT) is an instrument used for transmitting linear position measurements electrically to a remotely located indicating or controlling device (Figure A-87).

The LMT is a transformer type unit composed of three basic parts: a winding assembly containing primary and secondary coils wound around a tubular center; a cylindrical case which encloses the winding assembly; a probe incorporating the magnetic core. In a typical installation, the case containing the windings is mounted in a stationary position while the movable probe is connected to the component whose travel is to be monitored. Linear movement of the component is followed by the probe, hence axial displacement of the core through the windings.

In operation, the LMT produces a secondary output voltage proportional to the axial displacement of the core from a center or null position. Output voltage produced when the core is positioned to one side of null (center position) is 180 degrees out of phase with the voltage produced when the core is moved to the opposite side of null.

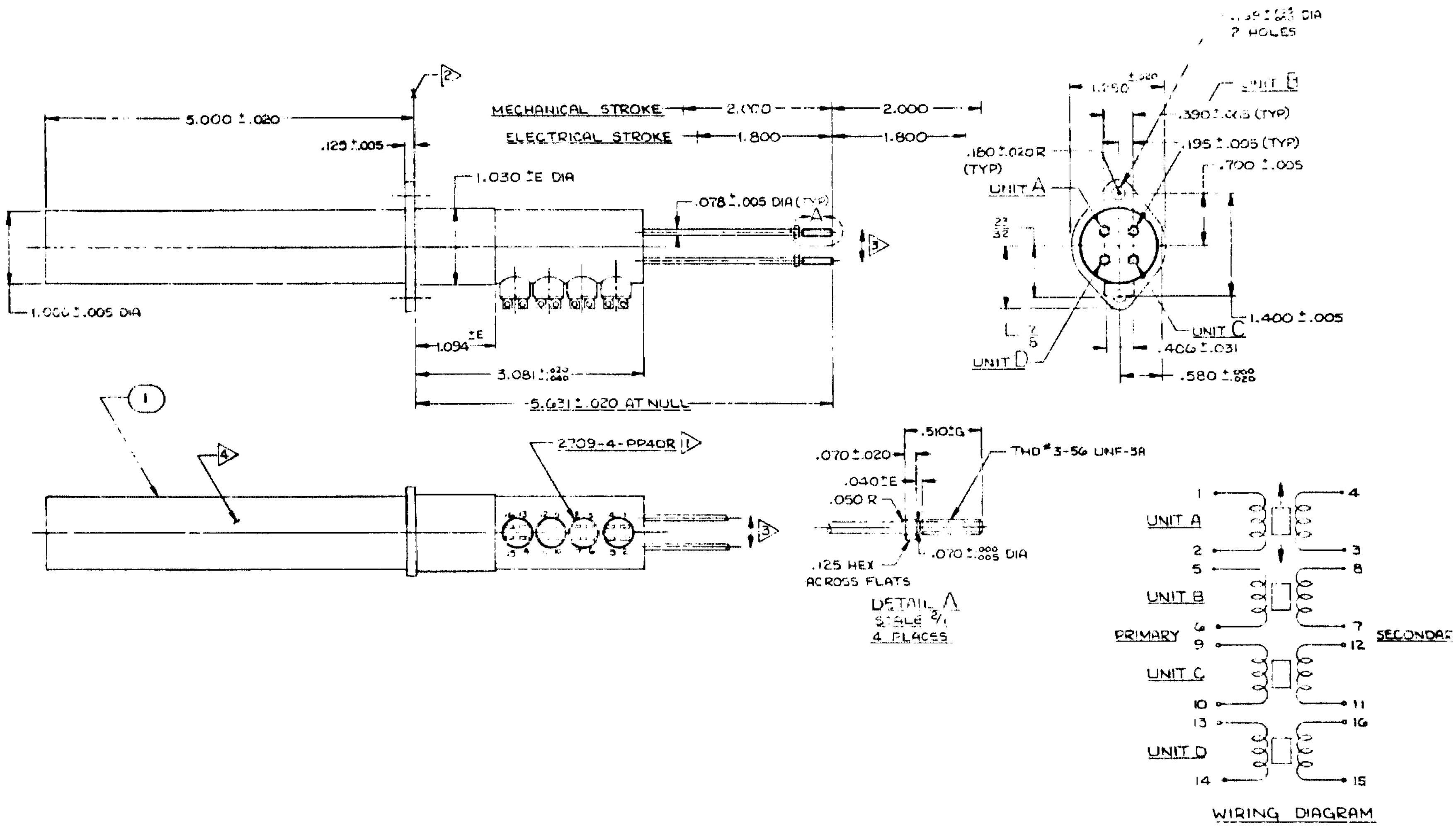


FIGURE A-87 LINEAR TRANSDUCER

The DC LVDT maintains all of the desirable characteristics of the AC LVDT, but has the simplicity of DC operation. It is comprised of two integral parts: the AC-operated LVDT and a carrier-signal conditioning module. Small, yet rugged, the carrier system eliminates the volume, weight, and cost of conventional external AC excitation, demodulation, and amplification equipment. The complete unit can operate from a simple power source, even dry cells. Virtually any DC meter can be employed as a readout.

Development of a practical DC-operated LVDT was not possible until the recent availability of miniature, high-performance, solid-state components. Early DC LVDT's, however, exhibited several undesirable characteristics. Their stability was poor and the output varied greatly. Figures A-88, A-89, and A-90 illustrate the new concepts.

- LOW-COST DC LVDT
- 24V DC POWER SOURCE OR PORTABLE BATTERY

The low-cost, HR-DC Series combines an HR Series AC LVDT with a discrete, solid-state modular signal conditioner. Silicon diodes and highly stable resistors assure optimum stability. The HR-DC's excitation frequency was selected to minimize sensitivity changes due to temperature fluctuations. The HR-DC can even be operated with a lightweight, portable battery because of its low current drain. Although the HR-DC has no amplifier, its output power is sufficient to operate most panel meters, as well as high input-impedance recording and control equipment.

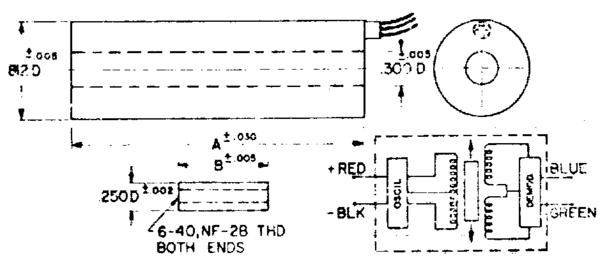


FIGURE A-88 DISCRETE ELECTRONIC COMPONENTS - HR-DC SERIES

- COMPUTER-DESIGNED FOR EXCEPTIONAL LINEARITY
- RESISTS SHOCK AND VIBRATION

The DC-D Series combines a hybrid, thick-film circuit signal conditioner with a computer-designed AC LVDT. This results in an extremely reliable DC-operated position transducer. The DC-D is normally powered by a regulated ±15 VDC supply and converts core displacements into proportional outputs up to ± 10 VDC. Microminiature components used in the construction of DC-D's are selected for maximum stability, and skilled assemblers bond the components to a ceramic substrate. All electrical connections are hand-soldered with the aid of special tools. Vacuum encapsulation of all elements produces an assembly that is virtually indestructible when exposed to shock, vibration, and other forms of physical abuse. Double magnetic shielding provides protection against stray electrical fields.

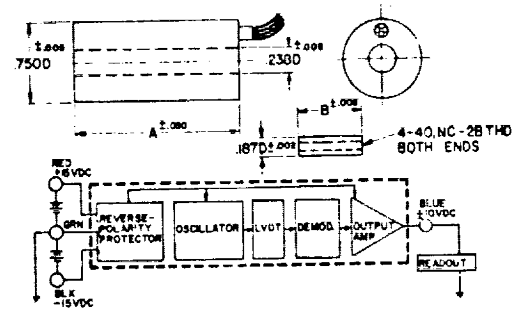


FIGURE A-89 HYBRID, THICK-FILM CIRCUITRY-DC-D SERIES

. HERMETICALLY SEALED BY TIG AND EB WELDING
. IMPERVIOUS TO HOSTILE ENVIRONMENTS

The HPD, HCD Series units are similar to the DC-D Series. Tungsten inert gas (TIG) and electron beam (EB) welding provide hermetic sealing that is free from oxidation-producing faults that may cause leakage. HPD and HCD units are impervious to dirt, water, steam spray, and corrosives. They have been qualified to 1000 psi and are suitable for numerous high-pressure applications. HCD units feature a glass-sealed, MS-type terminal connector, whereas HPD units employ a glass-sealed, pin-terminal header that allows the core and core rod to pass through the unit. Both HPD and HCD units have double magnetic shielding that makes them insensitive to external magnetic influences.

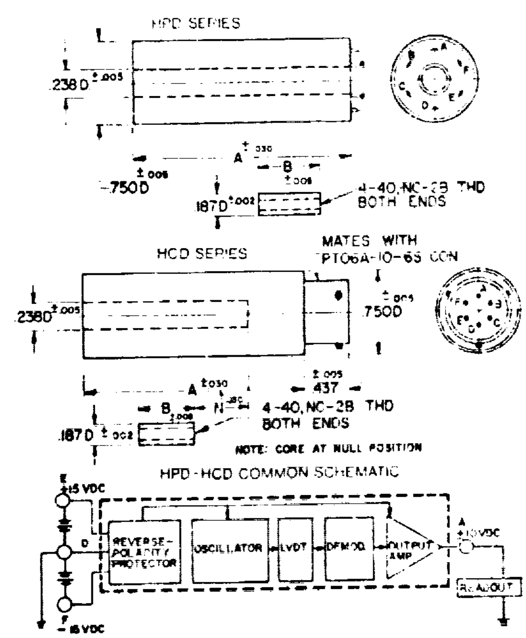


FIGURE A-90 HERMETICALLY SEALED - HPD, HCD SERIES

CONCEPT 63-C

ROTARY VARIABLE DIFFERENTIAL TRANSFORMERS

The Rotary Variable Differential Transformer (RVDT) is an angular position transducer (Figure A-91 and A-92) that retains the inherent precision of the LVDT. The RVDT's output is dependent upon AC induction variations as a function of angular position. This eliminates all moving contacts (such as brushes) because of the frictionless electromagnetic coupling.

Although the RVDT is capable of continuous rotation, most RVDT's operate within a range of ± 40 degrees. Within this range, linearity is better than ± 0.5 percent of full scale displacement. However, over smaller angular displacements linearity improves substantially. For example, the linearity for a displacement angle of ± 5 degrees is better than 0.1 percent full scale. The practical upper limit of an RVDT's angular displacement is approximately ± 60 degrees. Resolution, on the other hand, is essentially infinite. For small angular displacement, resolutions to a very small fraction of a degree are quite common.

In practice, only one of the two linear regions is calibrated at the factory. The null position corresponding to this factory-calibrated region is identified by appropriate marks on both the body and shaft. As with the LVDT, the RVDT's output voltage characteristic shifts 180 degrees in phase around a null or zero angle shaft position.

The AC-type RVDT (R30A) requires an AC voltage to energize its primary coil. It produces an AC voltage from its secondary (output) coils directly proportional to the shaft position.

The DC-type RVDT (R30D) accepts a DC input voltage that is internally converted to an AC carrier for exciting the primary coil. An integral demodulator and filter convert the secondary coil voltage into a smooth DC output signal which is subsequently amplified. Thick-film circuitry is utilized throughout.

Operating shafts on both AC and DC models are supported by miniature precision ball bearings, minimizing hysteresis and friction. The external housing is constructed of anodized aluminum, and internal shielding provides electrostatic and electromagnetic protection.

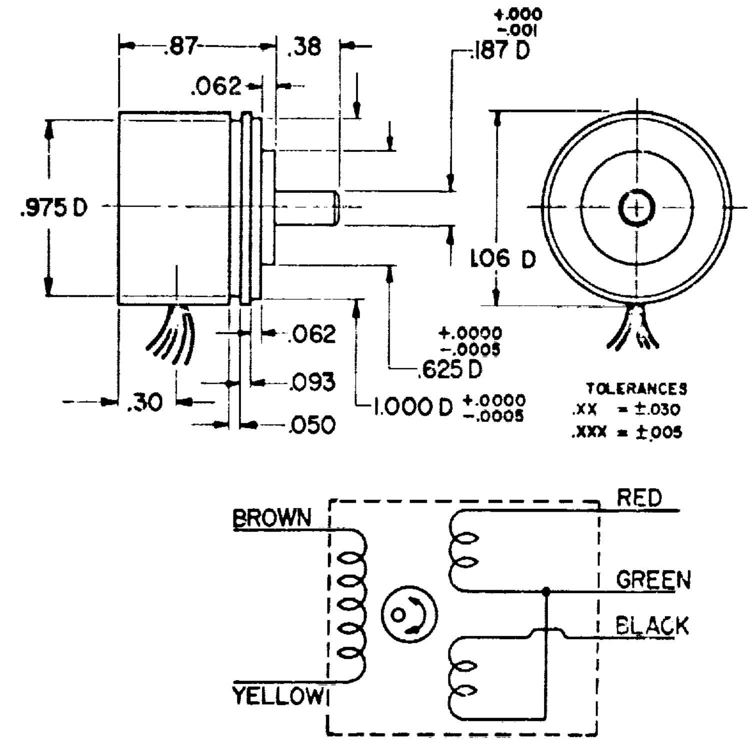


FIGURE A-91 AC RVDT

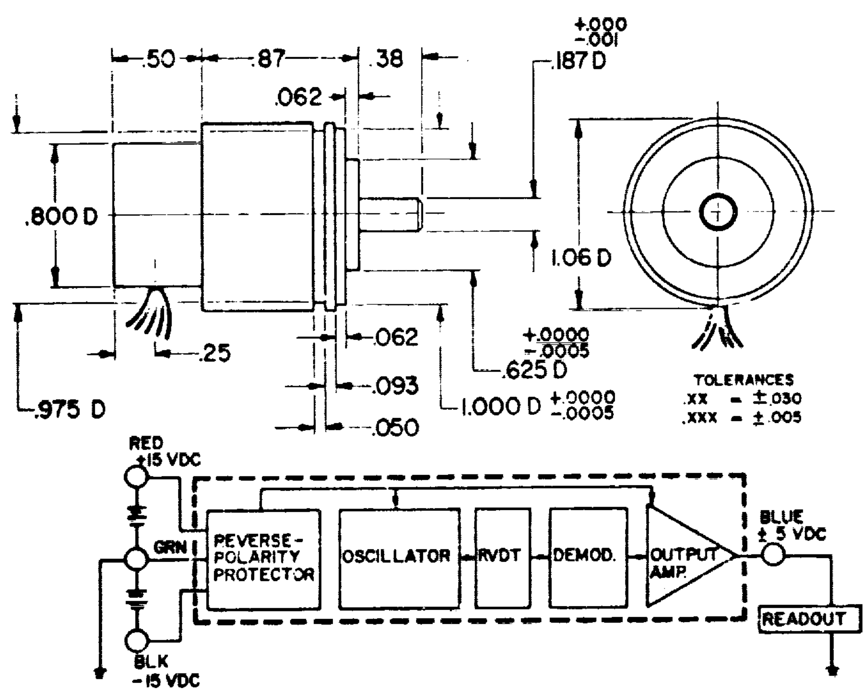


FIGURE A-92 DC RVDT

The Datex contacting encoders incorporate the most recent state-of-the-art advances in brush encoder design, including EMI suppression circuitry, internal transistor buffers for special load conditions, and high transition accuracy. (Figure A-93).

The encoders provide parallel contact closure outputs, usually designed to have continuity for a logic "1" and open circuit for a logic "0". The contact ratings for the standard models are 20 μa-3 ma per bit at voltages up to 15 vDC. Special loading conditions can be accommodated, however, by the use of internal buffers.

To increase the usable life of the units and accommodate special requirements, Datex has designed a number of standard electronic circuits which can be used in the encoders to benefit the user. These circuits provide for special load requirements, contact noise suppression, code translation and EMI suppression. In all cases, this circuitry is internal to the encoder and eliminates the need for the user to design special circuits in his system.

This Datex series of encoders has been designed and tested to meet all of the normally applicable military specifications, including MIL-E-5400, MIL-E-5272C, MIL-E-16400, MIL-I-16910, MIL-I-26600 and MIL-I-6181D. Brush redundancy is used in all models to increase reliability and insure longer encoder life.

Encoder Type	Overall Length
Single Stage	1.000 max.
Two Stage	1.500 max.

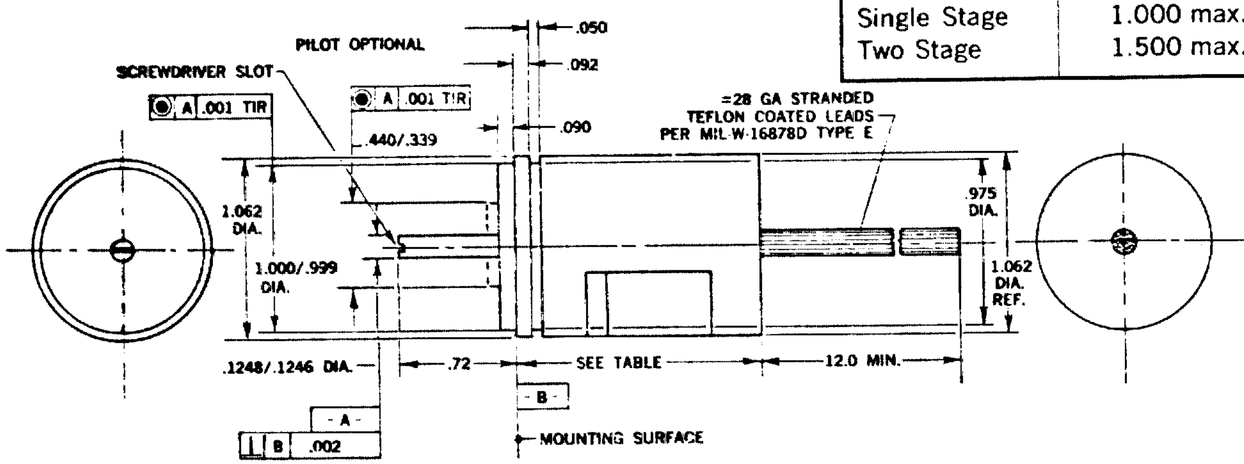


FIGURE A-93 CONTACTING SHAFT ENCODER

CONCEPT 65 -C OPTICAL SHAFT POSITION ENCODERS

These Datex encoders, designed for rugged industrial applications, include both single and multi-turn versions. In the multi-turn models, internal step-down gearboxes are used to provide as many as 134,217,728 counts over 65,536 turns of the input shaft (Figure A-94).

All major output codes are available — Gray, natural binary, Datex, BCD. All outputs are completely non-ambiguous, even in the multi-turn models. Ambiguities are eliminated by the use of special code patterns and electronics within the encoder. Resolutions per turn of up to 2,048 counts (.1758°) are available.

The electrical features of the Datex industrial encoders include either integrated circuit logic levels or discrete component logic levels; requirement for only two power inputs, which do not require precision regulation; ability to have continually following and interrogation modes of operation. Each encoder in both the single and multi-turn versions contains its own circuit card, which houses the shaping circuits, amplifiers, and logic circuits. Additionally, all models feature pre-focused lamp assemblies which are field replaceable in minutes.

Construction features of the Datex industrial encoder series include heavy aluminum housings, stainless steel shafts, sealed external bearings, shielded internal bearings, glass epoxy circuit cards. All bearings are rated to withstand 25 in./lbs. of bending torque. External surfaces are principally black-anodized.

Model	Encoder Length	
	Dim "A"	Dim "B"
05-509	3.35	3.00
15-509	5.25	4.90
25-509	7.50	7.15

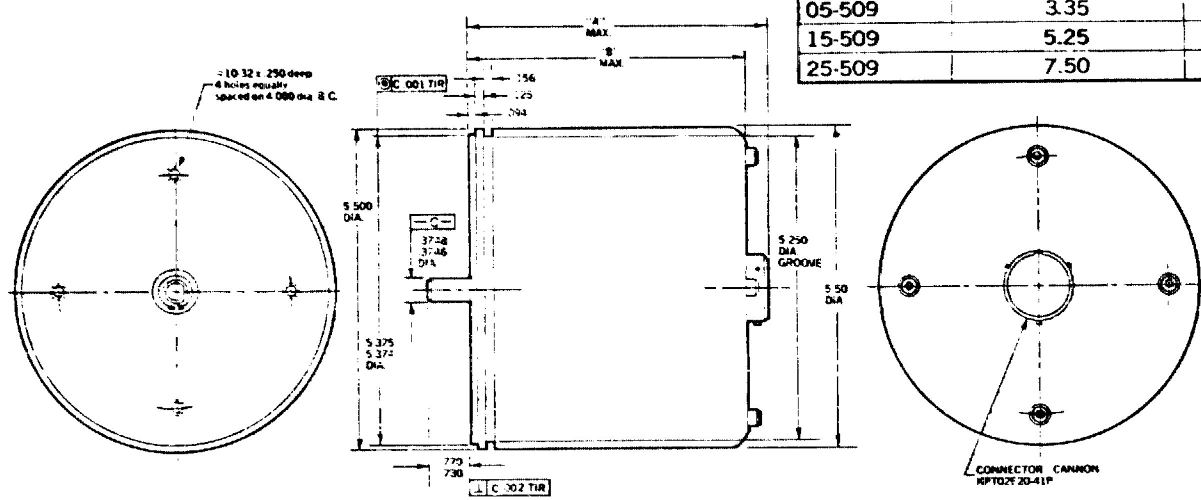


FIGURE A-94 OPTICAL SHAFT ENCODER

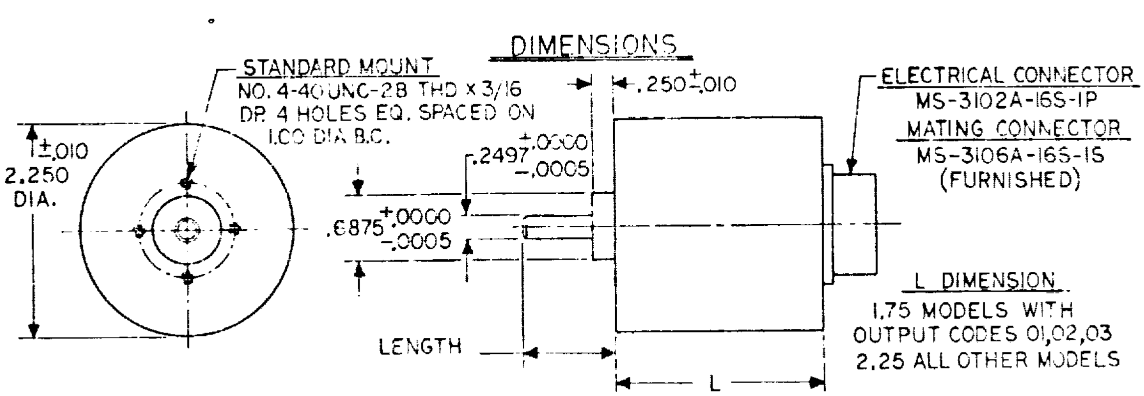
CONCEPT 66-C OPTICAL INCREMENTAL SOLID STATE ENCODER

Conrac Incremental Encoders are used as pulse generators for shaft speed monitoring or control, or as shaft position indicating devices when coupled with suitable counters. These instruments are designed specifically for commercial applications, with the ability to provide trouble-free performance under difficult environmental conditions. The light sources are gallium arsenide LED's, with a minimum of 100,000 hours of useful life. (Figure A-95).

Single channel or two channel outputs with 90° quadrature are available. Direction sensing can be provided on two channel models. The optional zero pulse output can reset counters at a selected shaft position, eliminating accumulated errors. Output levels are optional, with direct sensor output or DTL/TTL compatible output available.

One cycle consists of one "on" and one "off" segment. Direct sensor output is low level, with no conditioning.

Pulse outputs are produced at the leading edge or the leading and trailing edges of the output from a single sensor for x1 and x2 outputs, with x2 providing two pulses per cycle. Combining x2 outputs from two sensors at 90 electrical degrees quadrature produces x4 outputs with four pulses per cycle. The zero reference appears once for each shaft revolution and consists of a long duration pulse in modes 01, 02, or 03, and as a 3 usec pulse in all other modes.



PIN DESIGNATION

FUNCTION	PIN
OUTPUT A	A
OUTPUT B	B
− 5VDC	C
+ 5VDC	D
SPARE	E
ZERO REF.	F
GROUND	G

FIGURE A-95 OPTICAL INCREMENTAL SOLID STATE ENCODER

CONCEPT 67-C

LOW TORQUE POTENTIOMETERS

Microtorque and Minitorque potentiometers are designed for applications where only extremely low torque loads can be tolerated. Such applications include the transmission of readings from gauges located in remote or hazardous areas, the indication of vane angle in aircraft pitch and yaw assemblies and of shaft angles in mechanical computer systems, the function of a transmitter in a Selsyn system, the measurement of motion or displacement in laboratory experiments, the indication of shaft position or wind direction indicators, and many others (Figure A-96).

The Model 85111 Microtorque potentiometer has jewel bearings and a precious metal winding. The operating torque is extremely low, with a maximum of 0.006 inch-ounces. Single or double brush construction is available; shaft diameter is 0.031 inches.

The Model 85151 Minitorque potentiometer uses sleeve bearings, a 0.125 inch diameter shaft and precious metal windings. Construction is more rugged than for the Microtorque and operating torque is higher; 0.025 inch-ounces.

The Model 85153 Minitorque is essentially the same potentiometer as the 85151, with ball bearings instead of sleeve bearings for lower operating torque; 0.010 inch-ounces. The shaft size remains 0.125 inches diameter and the same precious metal windings are used.

All models feature stainless steel shafts, plastic bodies and anodized aluminum covers. Weight is a low 0.6 ounce maximum.

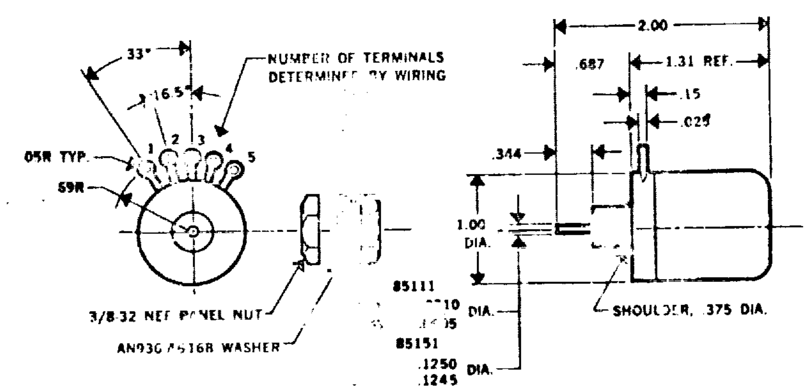


FIGURE A-96 CONRAC LOW TORQUE POTENTIOMETERS

Both binary and BCD types use a solid gold interrogation track. Voltage applied to this track provides for NOR logic interrogation, while grounding it changes brush selection to permit NAND logic interrogation. Binary encoders use a V-scan type of brush selection; BCD types employ a V- and U-scan combination. See Figures A-97, A-98 and A-99.

Using proper V-scan selection logic, binary encoders provide a non-ambiguous parallel-binary output. Isolation diodes incorporated into these encoders permit encoder time-sharing in a specific system. Ambiguity cannot result from mechanical misalignment because the code pattern employed in any of these encoders is designed so that each brush selected for interrogation is never on or off its respective bit by more than ½ of that bit. This means that under any condition of dynamic, or "on-the-run" interrogation, the only bit changing its mechanical position is the one which is least significant. Even this change is not electrically sensed by the encoder, for a flip-flop used on the least significant bit maintains its state properly at the instant of interrogation. Input shaft speed determines the interrogation time required to assure position change in none but the least significant bit.

Output of BCD (8,4,2,1) encoders is serial, in contrast to the parallel output of pure binary types. Though output is serial for a total reading, each digit is parallel in form through use of a logic package permitting digital time-sharing. If time-sharing is not required, then a parallel output BCD (8, 4, 2, 1) encoder can be used. A logic package, however, is required with each decade for parallel operation.

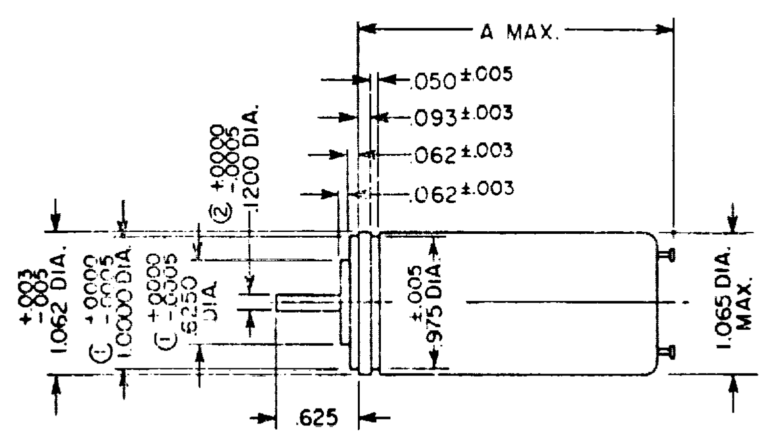


FIGURE A-97 KEARFOTT BRUSH ENCODER

V-scan selection logic, used on all Kearfott binary code converters, operates such that the state of the least significant bit determines selection of the next significant one. That one, in turn, determines selection of the next and so on until the last, or most significant bit has been read. Only one brush is needed for the 2^0 and 2^1 bits but two brushes - one lead and one lag - are used for each succeeding bit. Only one of these brushes can be selected at any particular instant.

By employing the V-scan selection technique, ambiguity of binary outputs is eliminated. In the illustration of the logic necessary to perform this function, circuit elements are identified as follows:

2^0 = Least significant bit (LSB)
2^1 = Next least significant bit
2^2 = Next least significant bit
2^N = Most significant bit (MSB)
D = Digit
C = Complement
▷ = Inverter
D = AND Gate
⊖ = OR Gate

This transistorized V-scan selection logic permits various output configurations. A logical "1" may be equal to 0, + V, or -V, while a logical "0" may be equal to 0, + V, or -V. Also, digit and complement are available simultaneously. In a system where time sharing of encoders is possible, one selection logic package of this type can be used to select the required number of encoders in the system.

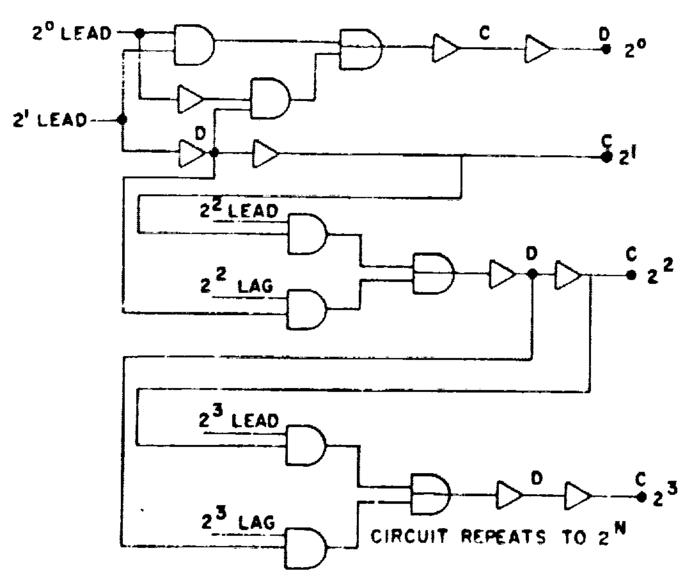


FIGURE A-98 V-SCAN ENCODER SELECTION LOGIC

BCD (8, 4, 2, 1) encoders, by combining V-scan and U-scan selection logic, provide outputs in serial form. The least significant four-bit digit is read first. Then the second, third, fourth, etc., digits are processed in sequence until the last digit is read. Each digit has its own common track and uses the same basic logic as that for binary encoders to present the first digit in parallel form. Using the same logic, the next digit is read, time-sharing the digits with one logic package. This provides a serial output for the total reading with a parallel output for each digit. E.G., a 3599 BCD encoder would be read 9 (1001), then 9 (1001), then 5 (0101), then 3 (0011).

If a parallel word output is required, the logic needed for one digit is repeated for each decade of the word, i.e., thelogic for one decade (9) in a 9999 BCD encoder must be repeated four times.

Plus and minus BCD encoders need a type of additional logic to generate the sign.

For parallel word output encoders, this circuit, less the flip-flop, is repeated for each decade:

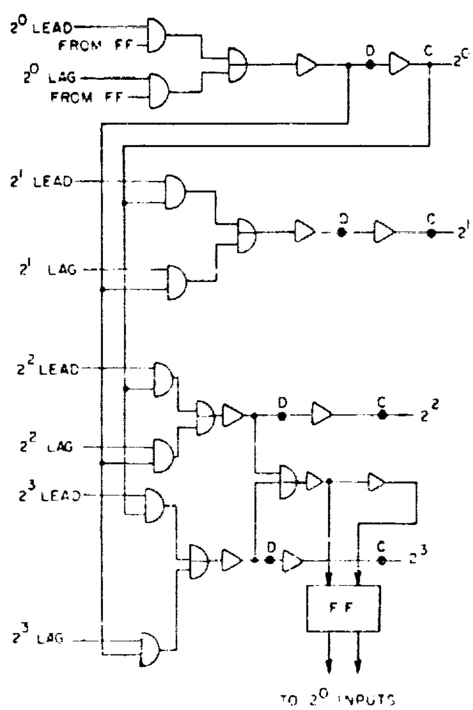


FIGURE A-99 V-SCAN/U-SCAN ENCODER SELECTION LOGIC

Figures A-100 thru A-106 illustrate some of the many types of synchros and resolvers currently available and their possible application to fly-by-wire flight control systems.

These units use rotary transformers to couple power into the synchro rotor in place of standard brushes and slip rings. Additional Kearfott features include the use of extra wide bearings for increased reliability and improved load carrying capacity. See Figure A-100.

This type of design provides system performance advantages in applications where synchros are driven at extremely high speeds or when brush wiping contact can not be allowed.

In comparison to standard brush type synchros, it should be understood that these type units do not provide complete electrical interchangeability. Use of rotary transformers reduces standard magnetic efficiency, increases phase shift, and increases the tranformation ratio temperature coefficient. However, in certain applications, particularly high speed, these units provide improved system performance.

Kearfott development programs are geared to maintain "state-of-the-art" quality, performance, reliability and lowest cost for both brush and brushless components.

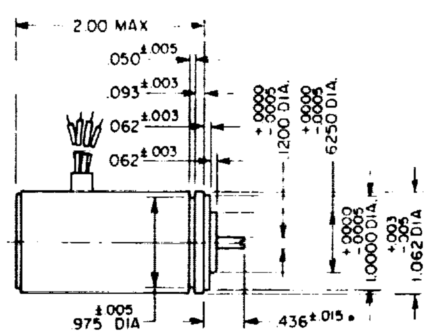


FIGURE A-100 BRUSHLESS SYNCHROS FOR CONTINUOUS ROTATION

These Kearfott units offer the same electrical characteristics as provided by standard brush type synchros. Their brushless construction is achieved by using spiral hair-spring conductors instead of conventional brushes and slip rings. See Figure A-101.

Since hairsprings require the use of mechanical stops to prevent "overwinding", Kearfott provides these units with uniquely-designed mechanical stops at angles such as 420° for applications requiring a full or more than 360° excursion, or at angles such as 340° for more limited rotation applications. In addition, other designs are available with stops at other angles depending on the user's particular application

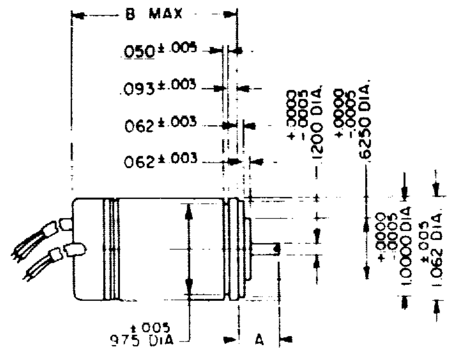


FIGURE A-101 BRUSHLESS SYNCHROS FOR LIMITED ROTATION

In construction, DC synchro repeaters consist of two orthogonal stator windings and a permanent magnet rotor. By energizing the two stator windings with DC voltages proportional to the sine and cosine, respectively, of an angle β , the rotor will orient itself with the resultant field at an equivalent angle θ . This device can be considered as the DC equivalent of a synchro torque repeater such as employed in various repeater indicators.

Other application advantages include: reduction in cost and size of drive circuitry compared to AC torque repeaters, no slip rings, and less power consumption required than AC torque repeaters.

In addition to the standard modifications of voltage ratings and shaft types, these units can also be provided with high torque gradients and special accuracy ratings. See Figure A-102.

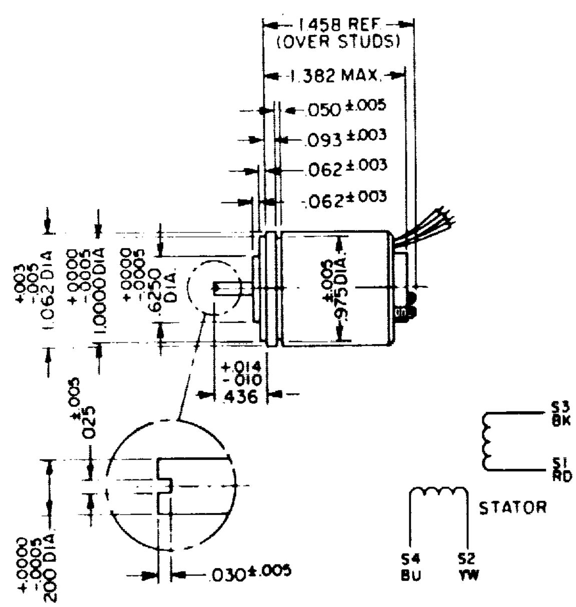


FIGURE A-102 DC SYNCHRO REPEATER

Tandem synchros provide two synchro functions in a single housing with a common output shaft. Tandem packaging eliminates synchro system gearing and associated backlash normally found when two synchros are coupled by 1:1 gearing. Kearfott produces a complete series of Size 8 and Size 11 synchros of which any two can be used in tandem, allowing the synchro user a wide selection of combinations depending upon his requirements. See Figure A-103.

Maximum electrical zero misalignment is held to three minutes on standard units. On request, Kearfott Size 8 and Size 11 tandem synchros are available having a maximum coupling error of only one minute. Size 8 tandem synchros in a single housing are 2.7 inches long and .750 inch in diameter. Length of Size 11 tandem assemblies is 3.7 inches for lead units and 3.9 inches for terminal units. Diameter of Size 11 units is 1.062 inches.

Also available is a design for tandem units consisting of two matched synchros mounted in a sleeve assembly. Size 8 sleeved units are 2.800 inches long and have a diameter of .795 inch and Size 11 sleeved units are 3.8 inches (max.) long and 1.125 inches (max.) in diameter.

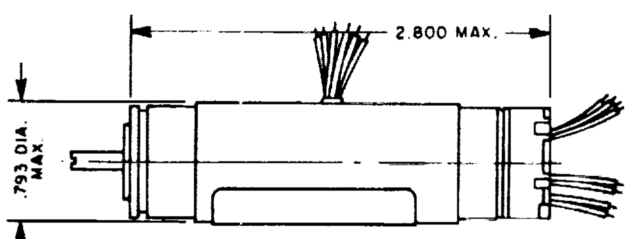


FIGURE A-103 TANDEM SYNCHROS

In general, the resolver consists of two primary and two secondary windings which can accommodate two inputs and two outputs. Since resolvers usually contain round rotors, there is no need to define two sets of impedance axes.

Usually the rotor-to-stator transformation ratio and phase shift are different from that observed in the stator-to-rotor direction. In torque and control-type synchros, where the primary and secondary elements are not arbitrary, the transfer function could be completely specified by input and output impedance and complex transformation ratio. However, to describe a specific resolver, the following terms are necessary: (1) rotor impedance; (2) stator impedance; (3) rotor-to-stator complex transformation ratio; (4) stator-to-rotor complex transformation ratio. See Figure A-104.

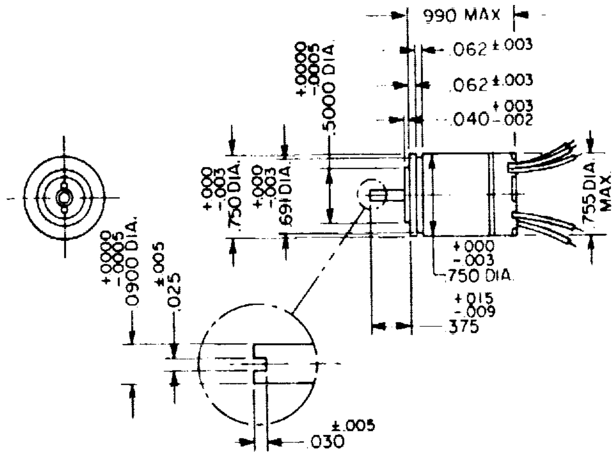


FIGURE A-104 RESOLVER

Winding compensated resolvers constitute a complete selection of components ranging from Size 8 to Size 25, and feature a wide variety of accuracies and impedances. They have been designed for high performance operation in computing chains, precise data transmission chains, analog computers, and automatic controls. See Figure A-105.

Stators of these components are wound with compensating windings which provide a feedback voltage for a buffer amplifier. Because the compensator and rotor (output) winding voltages vary with temperature and frequency in a parallel manner, the feedback loop automatically adjusts the transformation ratio and phase shift, holding variations to a minimum.

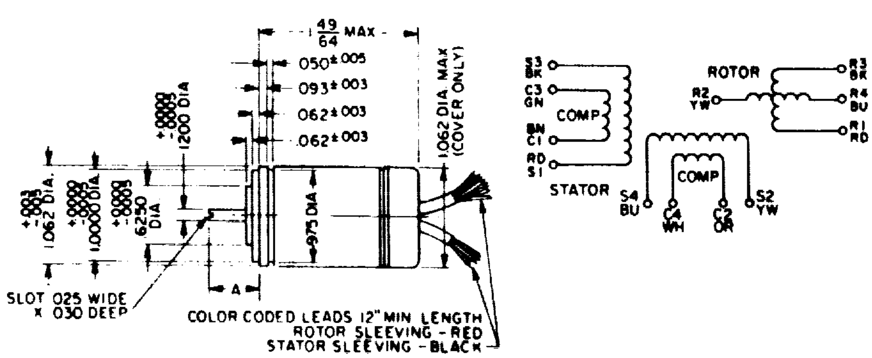


FIGURE A-105 WINDING COMPENSATED RESOLVERS

In addition to regular types of synchro control transformers, Kearfott designs and manufactures differential control transformers of which these units are representative types. These components, high precision units, are equipped with a rotatable stator at one end of which a gear is affixed. A small segment of this gear's circumference protrudes through a cutout in the housing to permit the engagement of an external gear train. By this means, two mechanical inputs are then made possible; one at the rotor shaft, and the other at the gear engaging that of the rotatable stator.

Output of these units takes the form of an AC voltage which is directly proportional to the sine of the algebraic sum of, or difference between the two mechanical input angles. See Figure A-106.

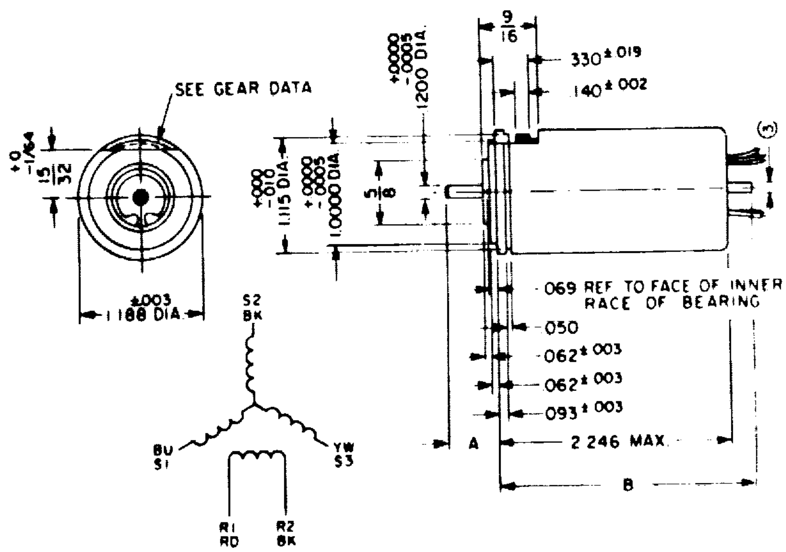


FIGURE A-106 DIFFERENTIALLY MOUNTED CONTROL TRANSFORMERS

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CONCEPT 70-C HALL EFFECT TRANSDUCERS

The classic hall effect, which is relatively immune to speed variations and relatively uncritical of the distance to the magnetic displacement medium, has been teamed with thin-film semi-conductor technology to produce a transducer that is suitable for applications like those show in Figure A-107.

This device can read a 1-mm magnetic track with a 3 VDC input, reading data recorded at 75 bits per inch, with a minimum output of 2 MV peak-to-peak. A typical device is shown in Figure A-107A.

Linear Displacement Transducer

The relative motion between a Hall device and a magnetic field produces a voltage which is a function of this motion. Permament magnets may be mounted as shown in Figure A-107B.

In Figure A-107BA the two magnets are aligned parallel to each other as shown. A zero plane, midway between the two N surfaces, will produce zero Hall voltage output when the Hall element is in the plane. Slight displacement to left or right will result in a large + or - output from the Hall device. This output voltage will indicate a very small movement and the direction of movement. In Figure A-107BB, a continuous field exists between the magnets but the field magnitude varies along a central plane. Movement of the Hall device in this plane, as shown, will produce approximately a linear output versus displacement.

Angular Displacement Transducer

The Hall voltage output is a function of the angle between the plane of the element and the direction of the magnetic lines of force. If this angle changes due to the rotation of the field or the Hall device relative to the other, then the output voltage is a sine function of this change. See Figure A-107C.

Various functions may be generated with the use of two Hall devices or by using a high gradient magnetic field.

CONCEPT 70 -C (Continued)

Proximity Detector or Brushless Encoder

Since one of the inputs to a Hall device is a magnetic field, this device is ideal for a non-contact proximity switch.

The switch may be designed to detect either the presence of a magnetic field or the disturbance of a magnetic field due to the presence of ferrous materials.

Figure A-107D shows the magnet in motion parallel to the Hall device and the resultant output voltage.

The use of flux concentrators would increase the sensitivity to a point where weak fields (such as the earth's field) could be detected. The magnet could be a permanent type or a magnetized spot on ferrous material.

By orienting the magnet as shown in Figure A-107E a null will be indicated at a precise point. This null can be used for positioning or to obtain accurate positioning information. This arrangement is particularly useful where the detection of both the presence and the direction of movement is desired.

Figure A-107F illustrates a ferrous material detector. It consists of a Hall device, a flux concentrator, and a magnet.

In Figure A-107F, the presence of the ferrous material increases the flux lines through the Hall device by decreasing the reluctance of the flux path.

By using similar methods, the sensitivity and resolution may be increased until movements and positions can be determined with .001".

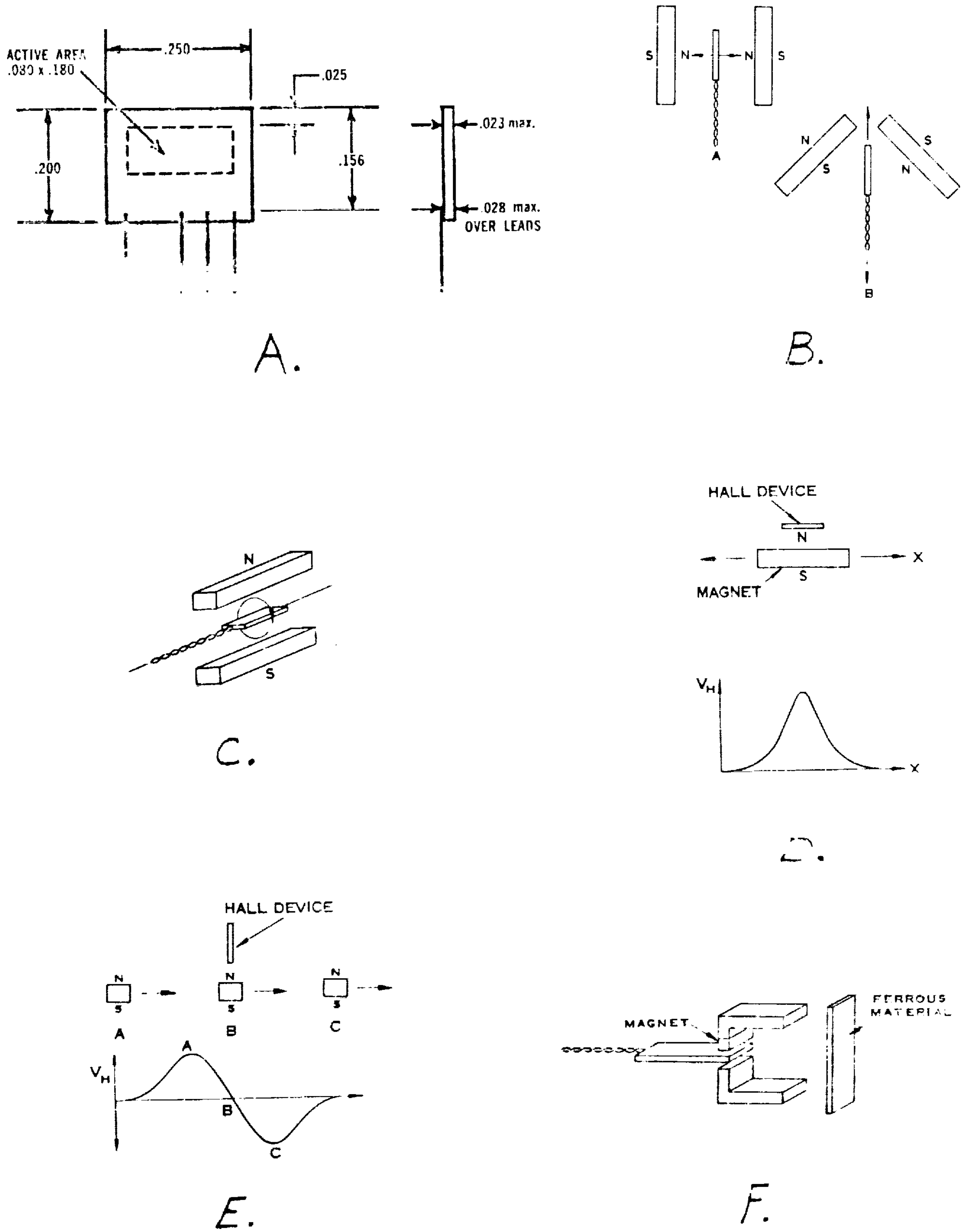

FIGURE A-107 HALL EFFECT TRANSDUCERS

CONCEPT 71-005 CH-46 HELICOPTER FBW SYSTEM

This conceptual fly-by-wire system could replace the current mechanical control system except for the cockpit controls and the feel system. This includes the following existing equipment: The control stick dual boost actuators (4), the series stability augmentation actuators (6), pitch series trim actuator (1), the mixing unit, the swash plate dual boost actuators (4), and all of the related interconnecting links. The fly-by-wire system replaces all of this equipment with stick position transducers (4), provisions for control signal shaping, mixing electronics, and swash plate actuators (4). The swash plate actuators combine all of the functions of the existing swash plate actuators, SAS actuators, and series trim actuators, and thus eliminate a considerable amount of complexity and weight. Eliminating the mechanical control linkages eliminates the need for the stick boost actuators. The electronics sum, shape, mix, and blend the stick position, SAS, and ASE (automatic stabilization equipment) signals to generate the proper control signals for the actuators. Various nonlinear or variable control functions can be inserted in the shaping electronics to evaluate their effects on control response or to tailor the response to the pilot's taste. The ASE signals interface mechanically with the cockpit controls through electromechanical servos to provide parallel stick motions. As an alternate configuration, the ASE could be interfaced electrically with the fly-by-wire system (as shown in Figure A-108), and thus further simplify the control system by eliminating the trim servos. Since trim motion feedback to the stick would also be eliminated by this approach, it is not recommended for this application. A simplified block diagram is shown in Figure A-108 mentation of quadruplex redundant electronics in the longitudinal axis is shown in Figure A-109. (The lateral axis would be similar.)

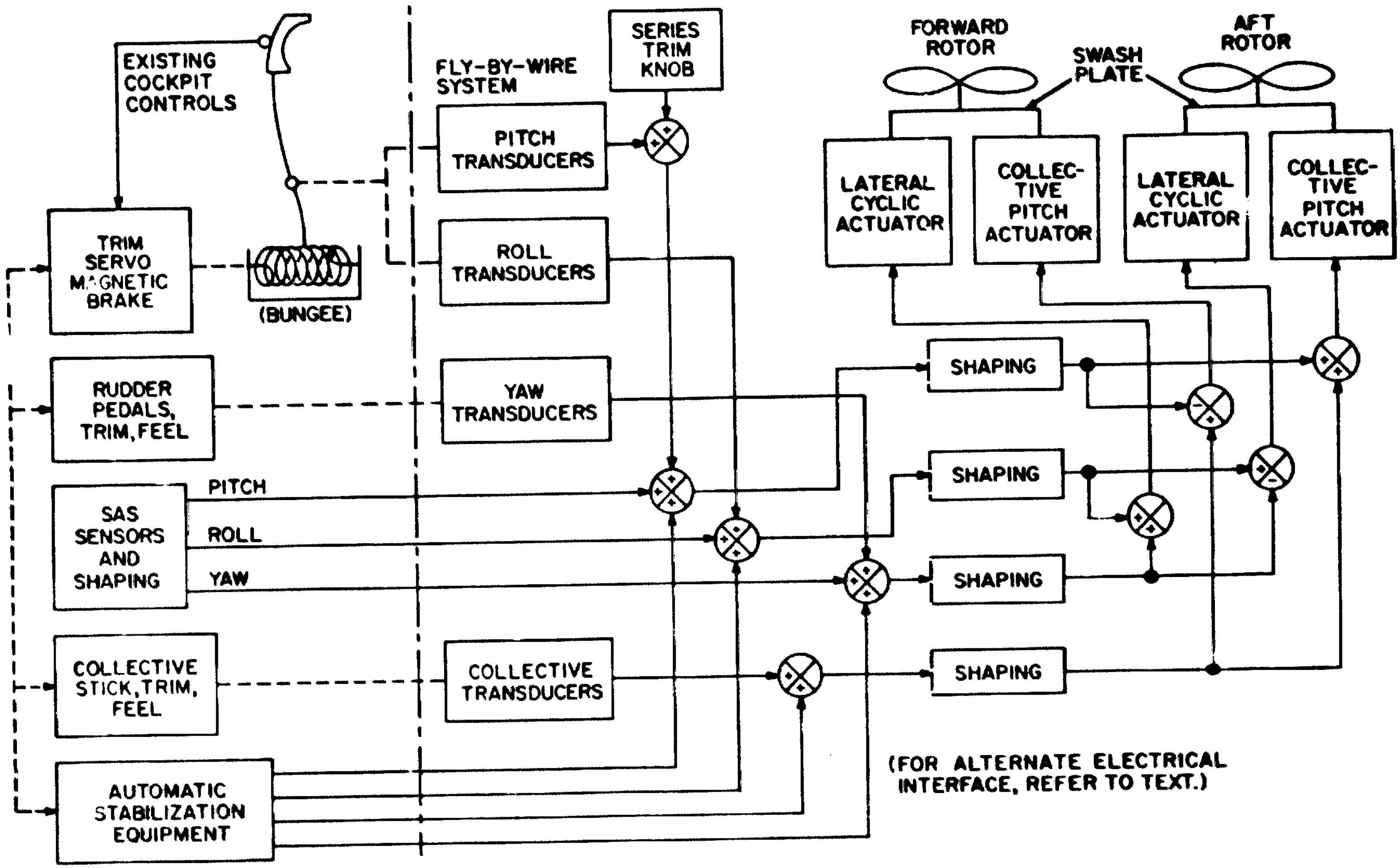

FIGURE A-108 CH-46 FLY-BY-WIRE SYSTEM - SIMPLIFIED BLOCK DIAGRAM

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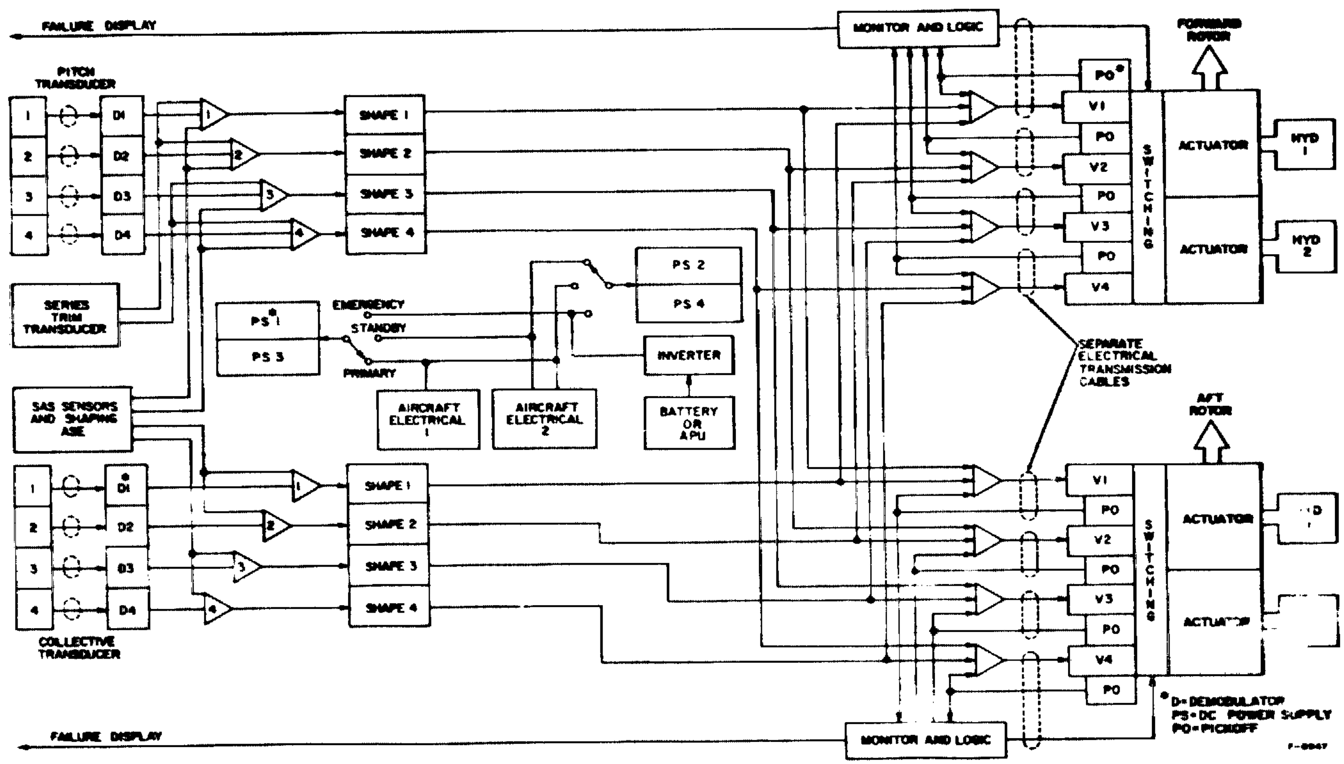

FIGURE A-109 FLY-BY-WIRE IMPLEMENTATION - LONGITUDINAL AXIS

CONCEPT 72-005 H-43B TWIN ROTOR HELICOPTER FLIGHT CONTROL SYSTEM

Kaman employed the H-43B twin-rotor helicopter for comparison of fly-by-wire and mechanical systems. Figures A-110 and A-111 show functional schematics of the derived electronic flight control system (EFCS) for the lateral cyclic and collective pitch axes. The lateral axis is independent of the other three so that the diagram shows the basic techniques derived. The collective axis combines the thrust (or lift) and directional axes, and the diagram shows the required interconnections. Figure A-110 shows that Kaman has used standby redundancy to achieve a fail-operational system. Triplex induction potentiometers serve as control stick transducers. One transducer provides a reference for comparison with the active transducer. When a failure occurs in either one, the monitor switches out the active unit and switches in the standby one. The system employs dual hydraulic actuators in standby redundancy. The fault detector (monitor) compares the rms values of the command and servo position transducer outputs to detect failures. Upon detecting a failure, the monitor switches out the active actuator and switches in the standby one.

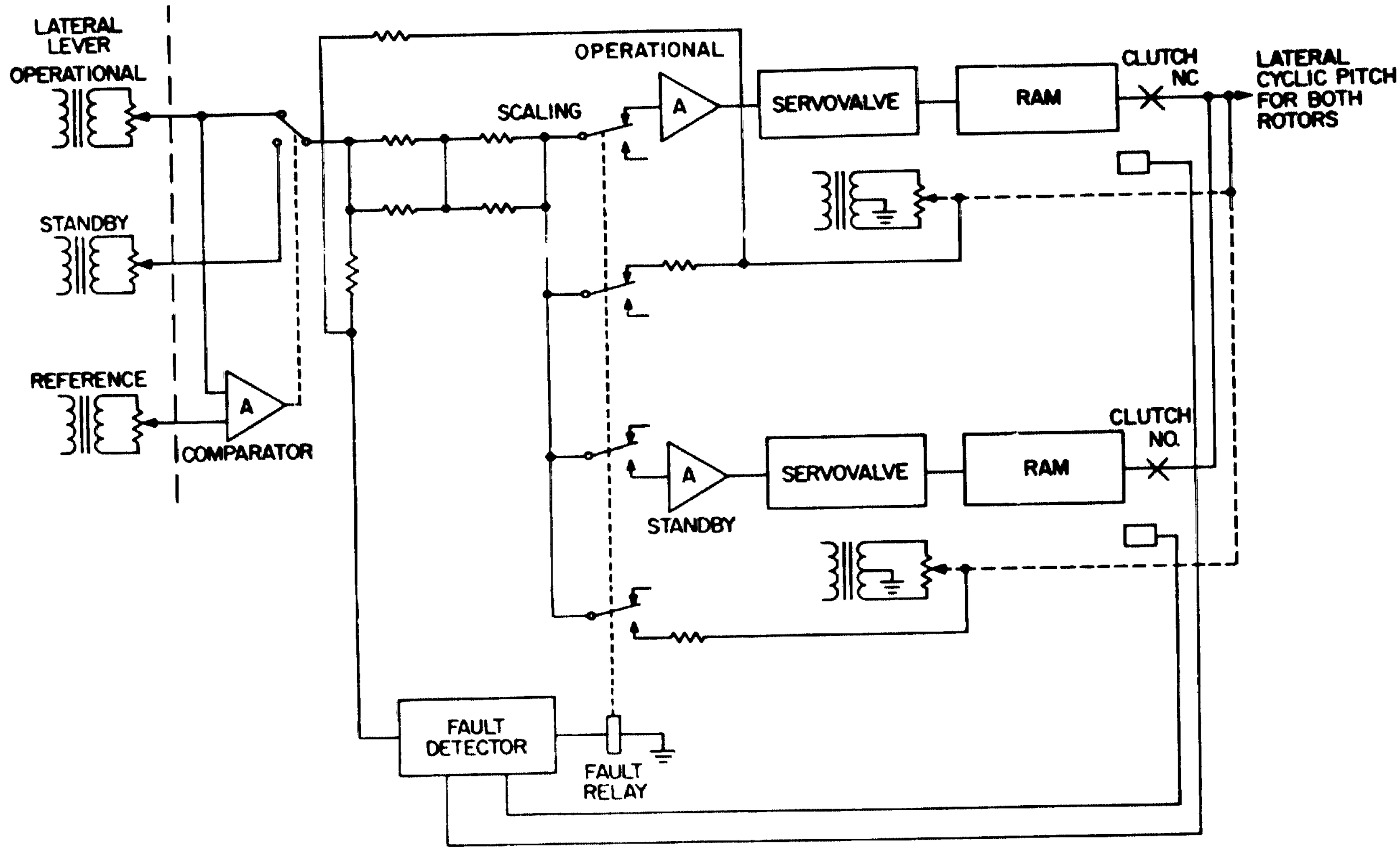

FIGURE A-110 KAMAN EFCS FUNCTIONAL SCHEMATIC, LATERAL CYCLIC AXIS

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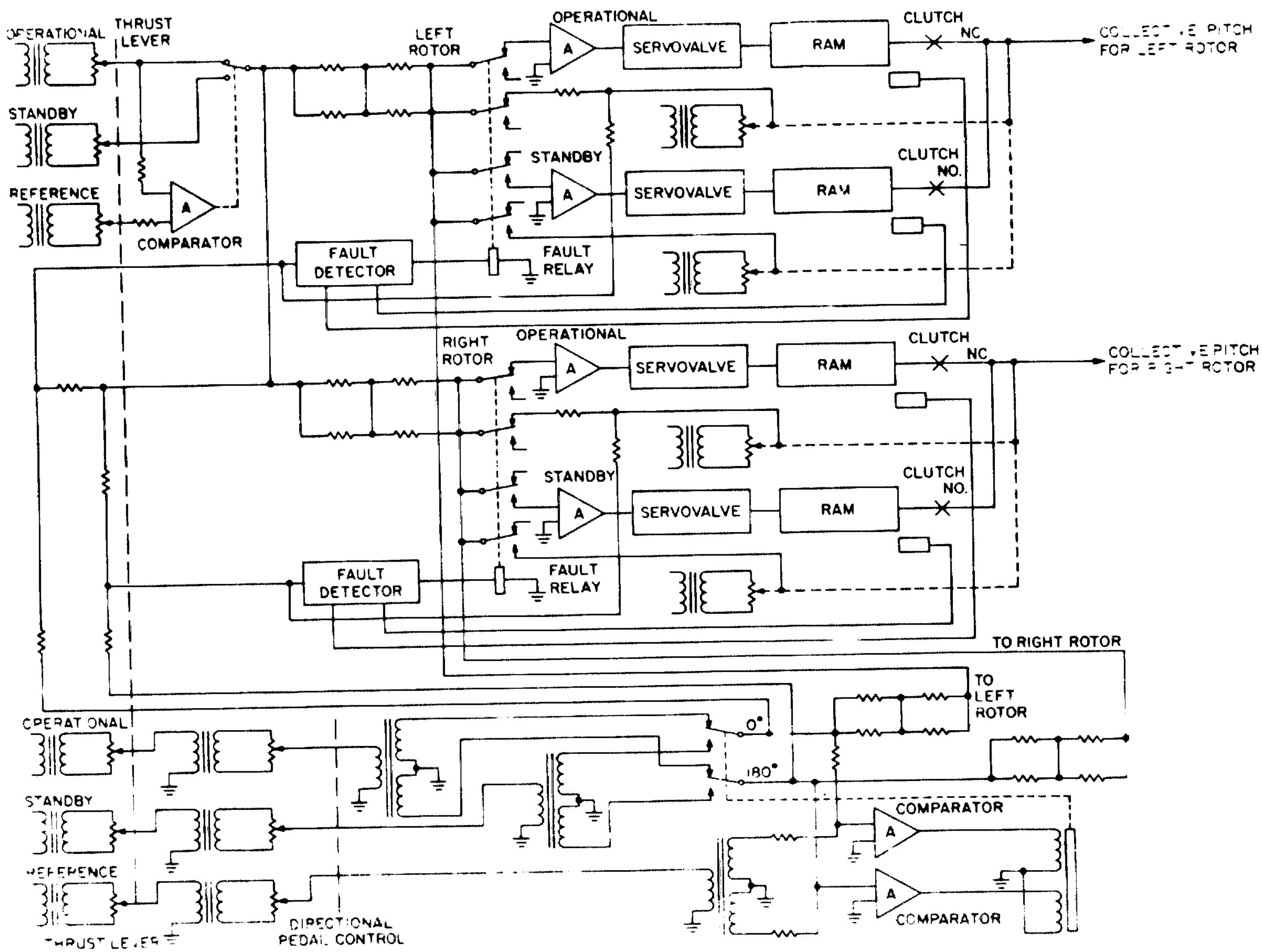

FIGURE A-111 KAMAN EFCS COLLECTIVE PITCH AXIS FUNCTIONAL SCHEMATIC

CONCEPT 73-005 X-20 DYNASOAR ELEVON CONTROL SYSTEM

The X-20 simulator, which uses most of the prototype hardware, is located at the Flight Dynamics Laboratory of the Research and Technology Division, Wright-Patterson Air Force Base. Figure A-112 shows a block diagram of the elevon control system. Most of the details of the aircraft are still classified, but conventional automatic flight control system design techiques were employed. The primary control system is fail-operational with an additional direct electrical link available for emergency backup control. The system is functionally very similar to the F-111 flight control system in that it uses essentially C* feedback and an adaptive gain control loop to maintain maximum servo gain and optimum handling characteristics. While the adaptive gain control is triplex in both systems, the remainder of the X-20 system differs in that it is only duplex. In-line monitoring and hard-over detectors continuously and independently check each channel to achieve the fail-operational capability. In case both channels fail, a direct link is available to provide a fixed surface deflection per stick deflection gradient. The backup link has no artificial feel, of course, and operators find flying it through the transonic range almost impossible.

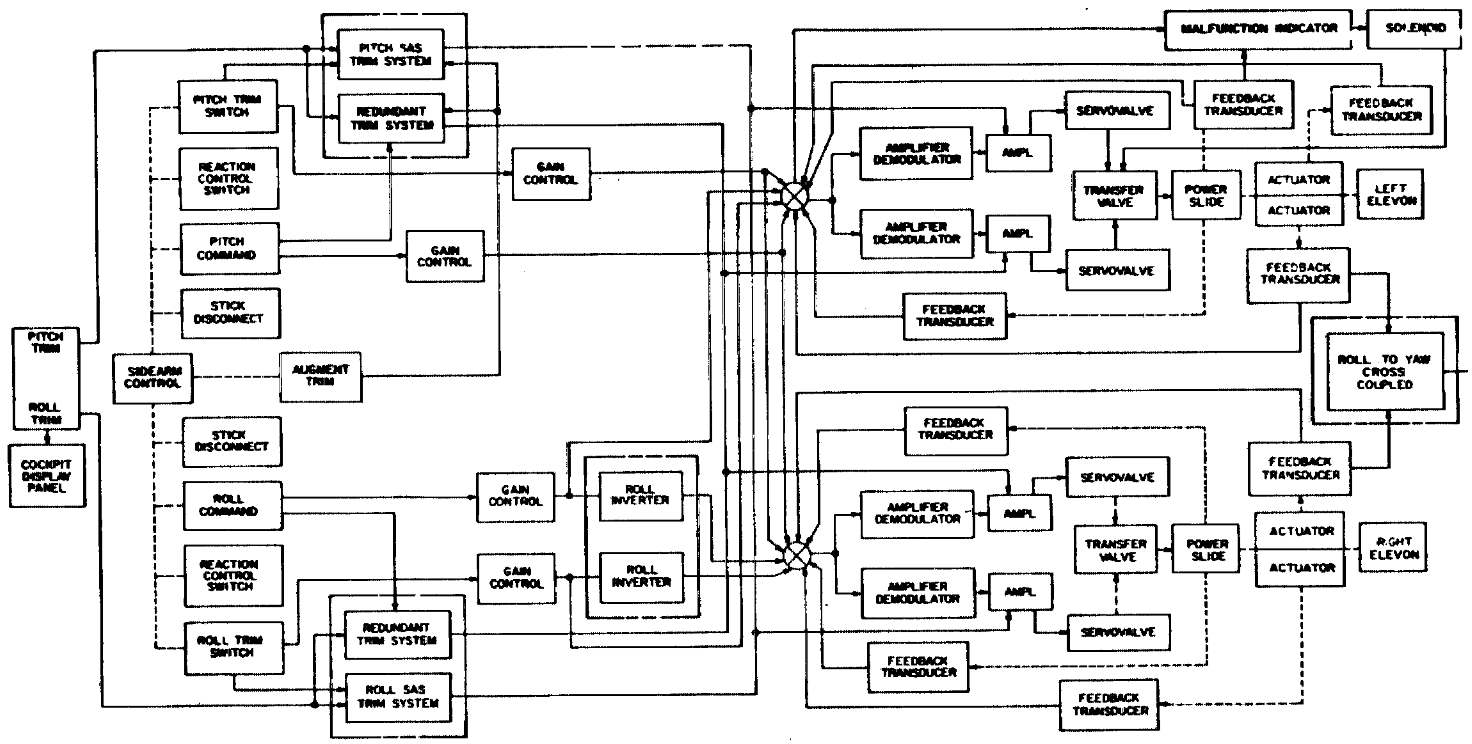

FIGURE A-112 ELEVON CONTROL SYSTEM BLOCK DIAGRAM

CONCEPT 74-005 F-111 A/B FLIGHT CONTROL SYSTEM

A fly-by-wire equivalent to the F-111 mechanical pitch control system is shown in Figure A-113. The system employs quadruplex position transducers to provide both command signals and actuator feedback signals. Left and right half redundant actuators are driven by quadruplex electronics and servo amplifiers. Stability augmentation, AFCS, series trim, and command augmentation signals are summed in the electronics and drive the redundant surface actuators. Parallel trim is still maintained to provide stick trim displacements if necessary. The basic components for implementing the fly-by-wire system are:

- Control stick
- Position tranducers
- Control electronics
- Parallel trim
- Feel spring
- Servo amplifier
- Surface actuators

The fly-by-wire system increases the cost of the pitch control system by 10 percent over that of the mechanical system. A weight reduction of 98 pounds and space savings of 400 cubic inches result by going to the fly-by-wire system.

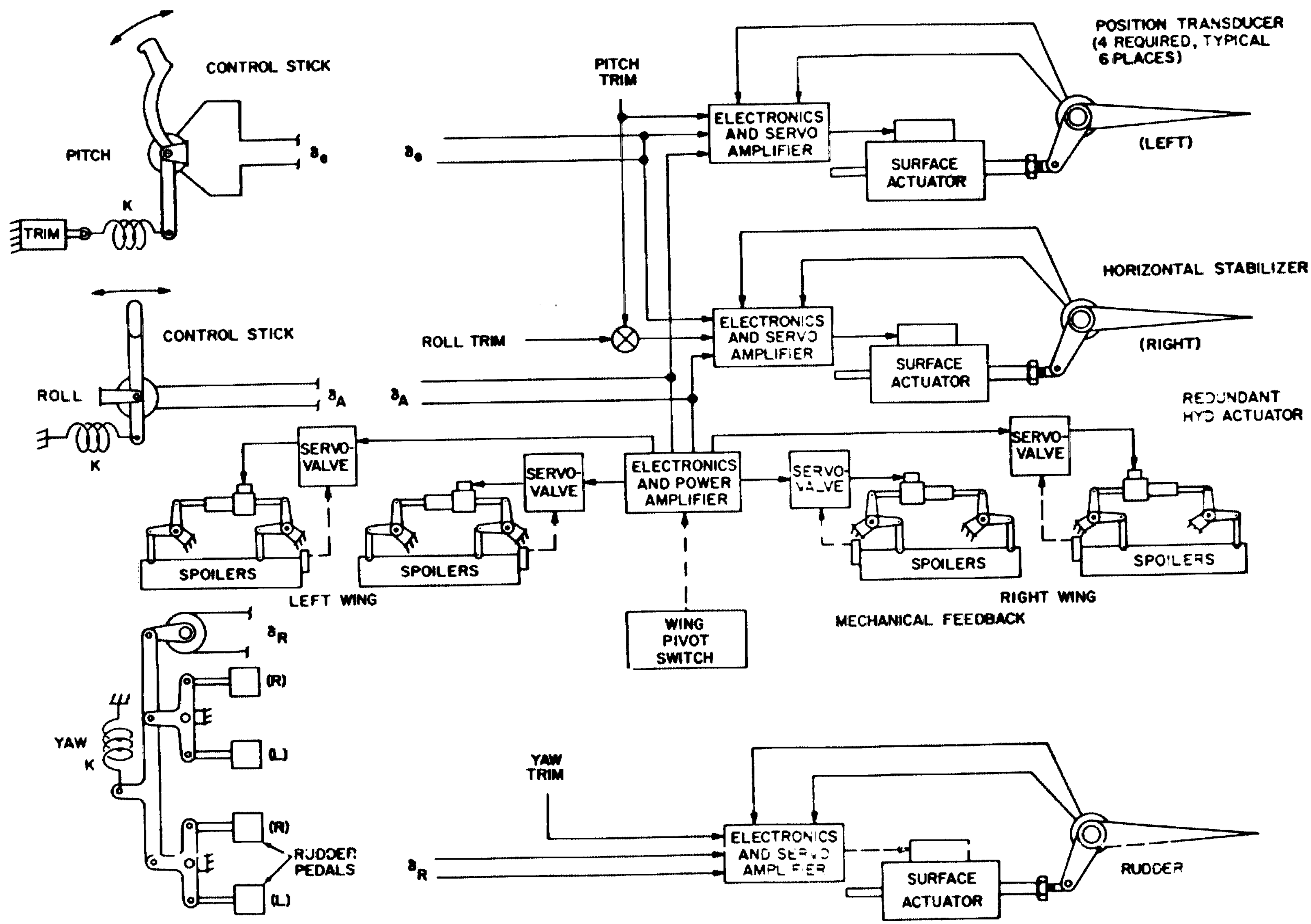

FIGURE A-113 F-111 A/B FLIGHT CONTROL SCHEMATIC

CONCEPT 75-043 ELECTRONIC FLIGHT CONTROL SYSTEMS

Mechanization diagrams of three basic configurations are shown in Figures A-114, A-115 and A-116. Redundancy levels are shown by the overlapping functional blocks. Interconnections between blocks indicate the general functional flow of information and not the number of interconnecting wires. The three-channel in-line-monitored configuration with the dissimilar backup channel is shown with a hydrofluidic backup channel in Figure A-116. Two backup configurations are a hydrofluidic configuration which will probably be more applicable to late generations of FBW systems, and an electronic backup channel which will be applicable to first-generation FBW systems. All three configurations provide dual fail-operation.

Referring to Figure A-114 the four-channel comparison-monitored configuration uses four identical rate sensor assemblies, four identical accelerometer assemblies, and four identical electronic control assemblies. It is assumed that these packages are spatially separated in the aircraft so that battle damage vulnerability is reduced, and chance of survival is increased. Each rate sensor assembly contains one each pitch, roll, and yaw rate sensor, and each acceleration sensor assembly contains one each normal and lateral accelerometer. Limited alignment equipment is assumed to be provided in the aircraft for initial installation alignment of sensor packages. It is considered advantageous to have all packages in the system alike for maintenance and logistics reasons. Only a single power input is required to each package, whereas in a four-channel, single-axis package, four input power leads are required to each package.

The electronic control assemblies each contain one channel of pitch, roll, and yaw command link and augmentation electronics, as well as the power supply, mode logic, and monitor and test circuitry for the system.

The stick force sensor assembly is a four-channel, two-axis assembly built as one unit to mount on the control stick. The requirement for physical separation of the redundant channels is not as stringent in the cockpit area, since it is assumed that the cockpit has some armor protection.

It does require that all four power sources be tied into this assembly. Because of space limitations on the control stick, the transducers are monitored in the computer assemblies.

The pedal position sensors are also assumed to be located in an armor-protected area of the cockpit and are considered to be mechanized into a redundant assembly. This will allow for calibration and alignment external to the aircraft before installation.

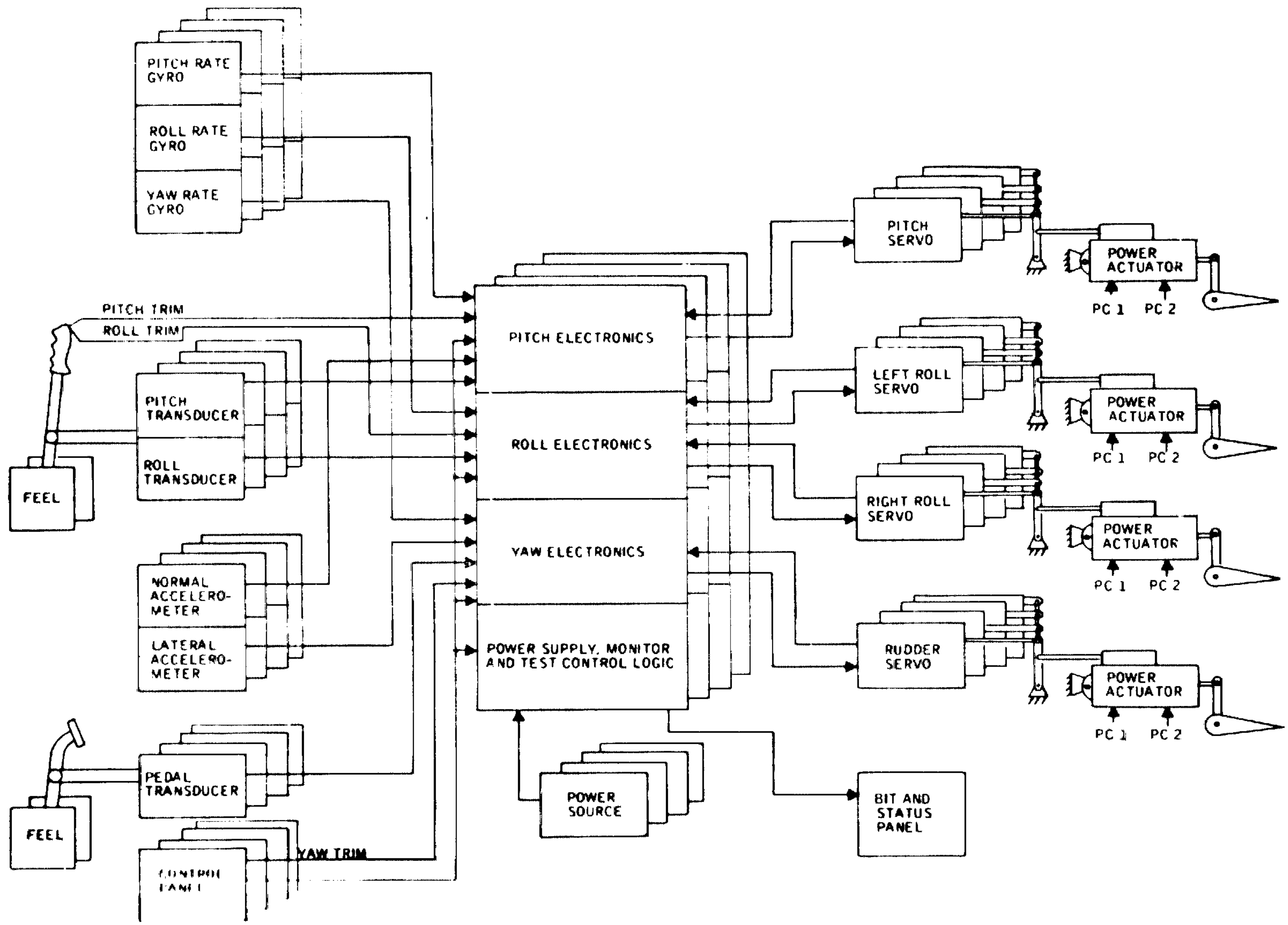

FIGURE A-114 FOUR CHANNEL COMPARISON MONITORED FLY-BY-WIRE PFC'S

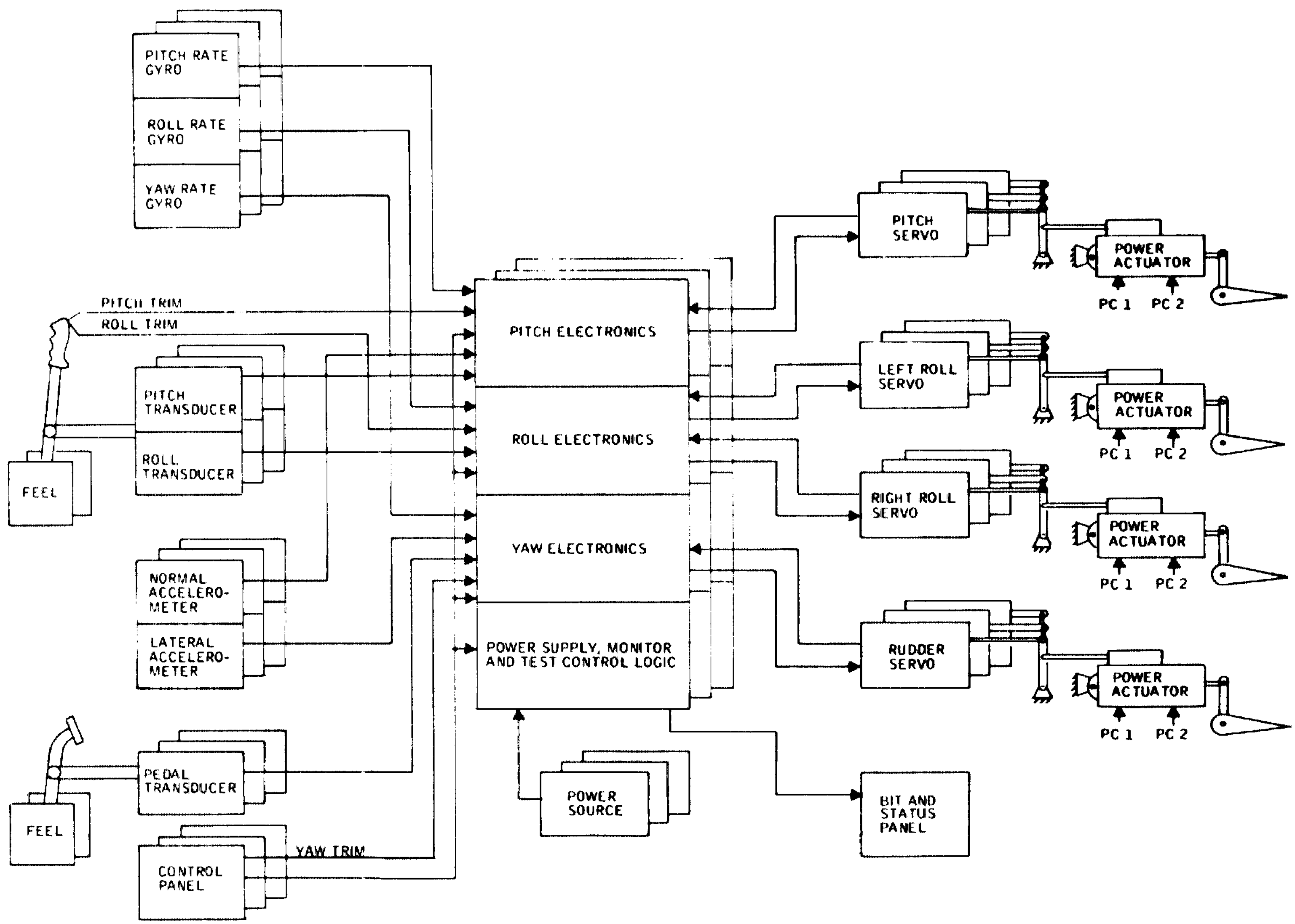

FIGURE A-115 THREE CHANNEL IN LINE MONITORED FLY-BY-WIRE PFC'S

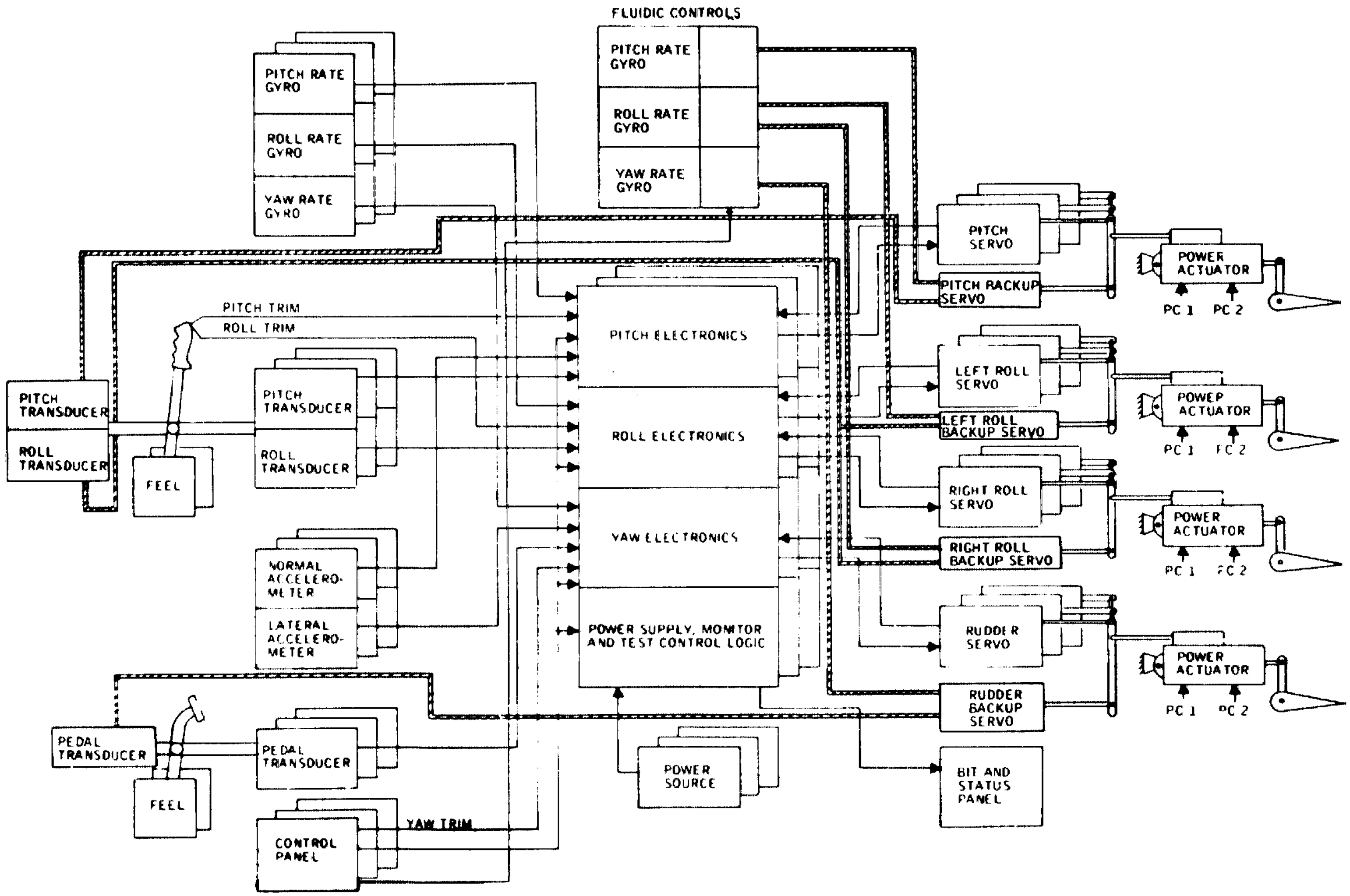

FIGURE A-116 THREE CHANNEL COMPARISON MONITORED FLY-BY-WIRE PFC'S

CONCEPT 76 -113 B-1 FLIGHT CONTROL SUBSYSTEMS

Horizontal Stabilizer Control

The pitch and roll SCAS is a dual-dual configuration which provides a fail operational failsafe system, Figure A-117. The pilot's and co-pilot's inputs are transmitted from the stick to the stick position transducers. The stick displacement signals from the transducers are gain scheduled as required by the SCAS controllers which command the SCAS servo actuators to displace proportionally to the stick. Air vehicle motion is sensed about the pitch axis by rate gyros and normal accelerometers whose signals are shaped and gain scheduled as required to control SCAS servo displacement proportionately to pitch rate and normal acceleration respectively. Feedback from the common and individual SCAS servo actuator position transducers close the control loop. Roll SCAS is augmented by the outboard spoilers at one-half flaps via the spoiler controller thus providing greater SCAS roll control at low speeds.

The SCAS has two pairs of two channels each. Each pair controls one of the dual tandem SCAS servo actuators and is self monitoring. If a failure occurs in one channel, the actuator is automatically centered and a warning light indicates to the crew a failure in SCAS. The results of a single failure in SCAS is the loss of redundancy only. The results of a double failure would be reversion to the mechanical systems with the SCAS inputs centered. This is indicated to the crew by a second warning light.

Spoiler Control

The electrical spoiler control provides roll control via the outboard spoilers and speed brake control, Figure A-118.

Electrical roll control is commanded by the pilot's and co-pilot's inputs to stick position transducers through the mechanical control path. The outputs of the position transducers are demodulated in the SCAS controllers and drive the spoiler controller. The output of the spoiler controller provides the electrical drive and control of the spoiler servo actuators. The electrical roll function is modified with stick deadband and mach lockout by the spoiler controller.

The mach lockout protects the wing structure from overload due to outboard spoilers. The stick deadband provides optimum roll control and aerodynamic efficiency.

Speed Brake Control

Speed brake control is commanded by the pilot or co-pilot by their inputs to the speed brake switches on the throttles and the motor select/emerg release switch. The inputs from the speed brake switches are channeled in EMUX to motor no. 1 or no. 2 by the motor select/emerg release switch modes of normal or alternate, and sent to the appropriate power relays which in turn power

CONCEPT 76-113 (Continued)

Speed Brake Control (Continued)

the speed brake actuator motors. If the emergency release mode is selected the input is sent from the switch through EMUX to the power relays which activates the dual emergency release mechanisms. The emergency release mechanisms disengages the speed brake actuator motors and allows the spoilers to close.

The output of the speed brake actuator drives the inboard spoilers and the position transducers. The position transducer outputs are transmitted via the SCAS controllers to the spoiler controller to activate the outboard spoilers speed brake function. The outboard spoiler speed brake function is inhibited in flight by the landing gear load switches whose inputs are transmitted to the spoiler controller via EMUX.

Rudder Control

The yaw SCAS is a dual-dual configuration, which provides a fail-operational, fail-safe system, Figure A-119. The pilot's and co-pilot's inputs are transmitted from the pedal position transducers. Pedal displacements are gain scheduled as required by the SCAS controllers, which command displacements of the yaw SCAS servo actuators to be proportional to the pedals. The common and individual SCAS servo actuators position transducers provide feedback to close the control loop.

The yaw SCAS has two pairs of two channels each. Each pair controls one of the yaw SCAS servo actuators and is self-monitoring. If a failure occurs in one channel, the actuator is automatically centered, and a warning light indicates to the crew a failure in yaw SCAS. The results of a single failure in SCAS is loss of redundancy only. The results of a double failure would be reversion to the mechanical system with the lower rudder centered. This condition is indicated to the crew by a second warning light.

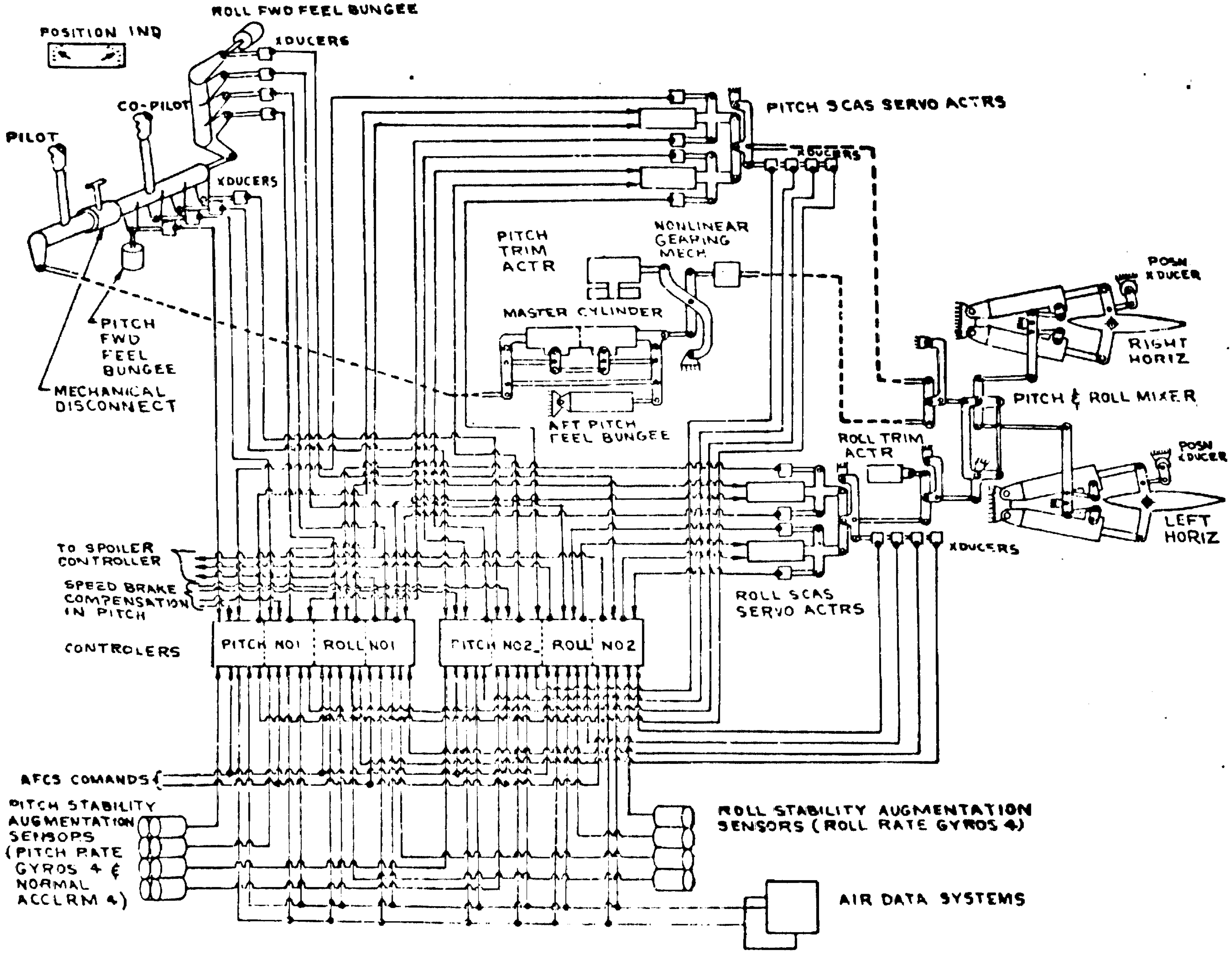

FIGURE A-117 HORIZONTAL STABILIZER CONTROL SCHEMATIC B-1

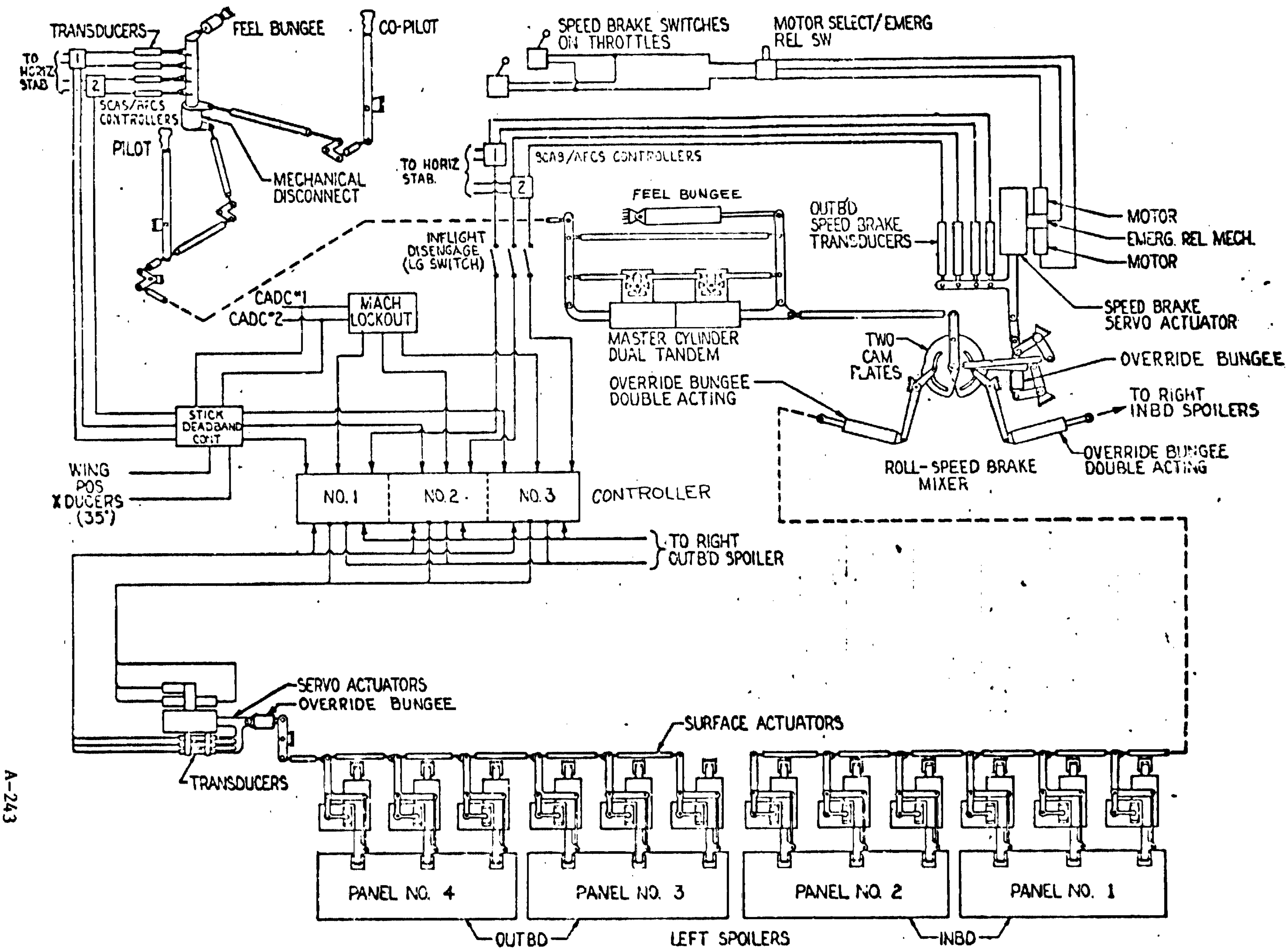

FIGURE A-118 SPOILER CONTROL SCHEMATIC B-1

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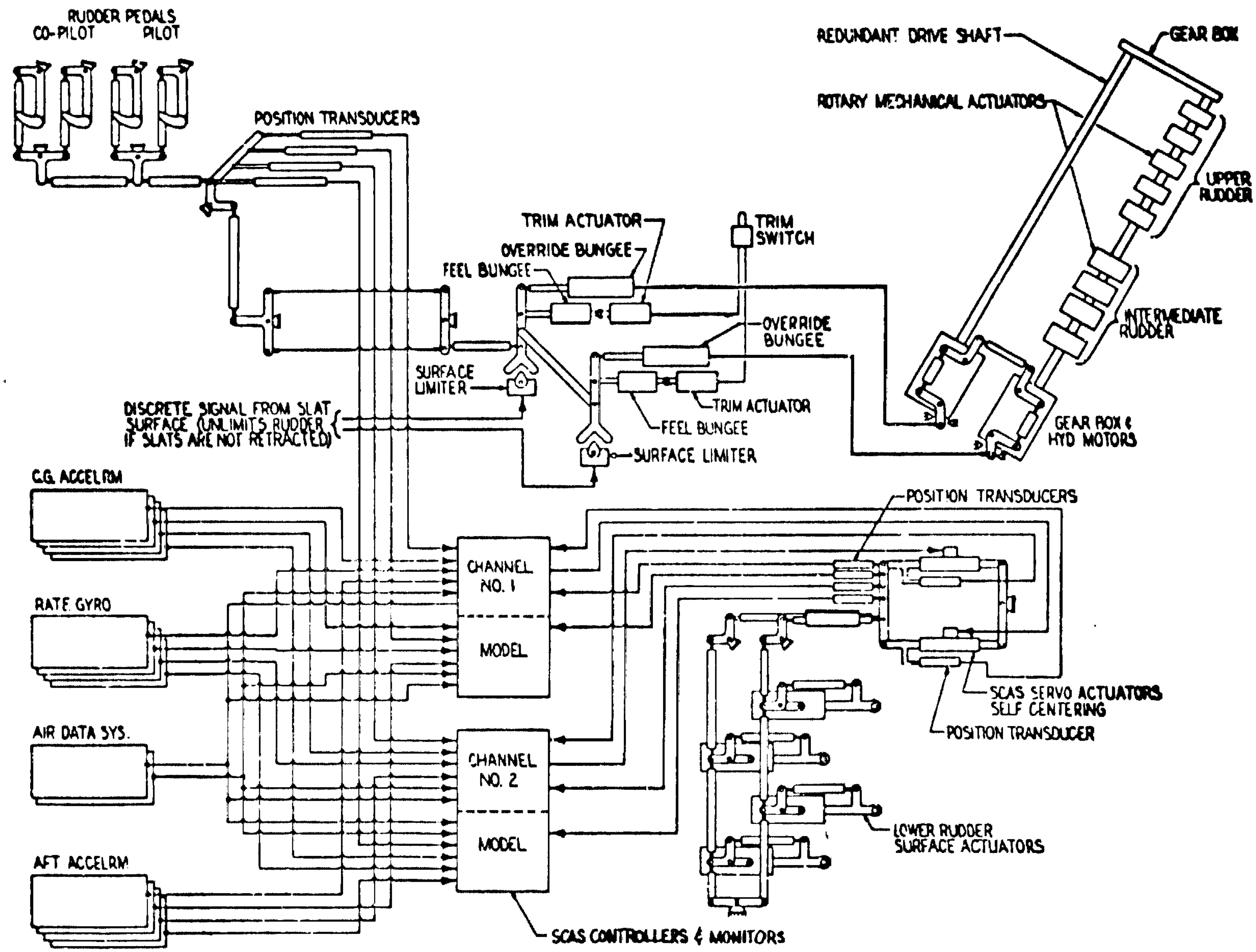

FIGURE A-119 RUDDER CONTROL SCHEMATIC B-1

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CONCEPT 77-160 HARRIER AUTOSTABILIZER SYSTEM

The inherent stability that can be achieved by aerodynamic effects with conventional aircraft is no longer available at speeds below the normal stall and while hovering.

A limited authority autostabilizer system is used in the Harrier. See Figure A-120. Within the autostabilizer computer, rate gyro signals are converted to signals proportional to rate and attitude with the attitude term leaked away to avoid saturation after pilot-demanded attitude changes. Pitch and roll channels are housed in a single 5½ lb unit containing gyros, computing, power supply and self-test channels. This unit is supplied by Elliott Flight Automation Ltd.

The combination of rate and sensitivity requirements dictated the use of particularly lively powered control units which are supplied for the Harrier by Fairey Hydraulics Ltd. Currently, position demand accuracy well within 0.001 in. (.025 mm.) is achieved. The valve flow/position characteristic is linear.

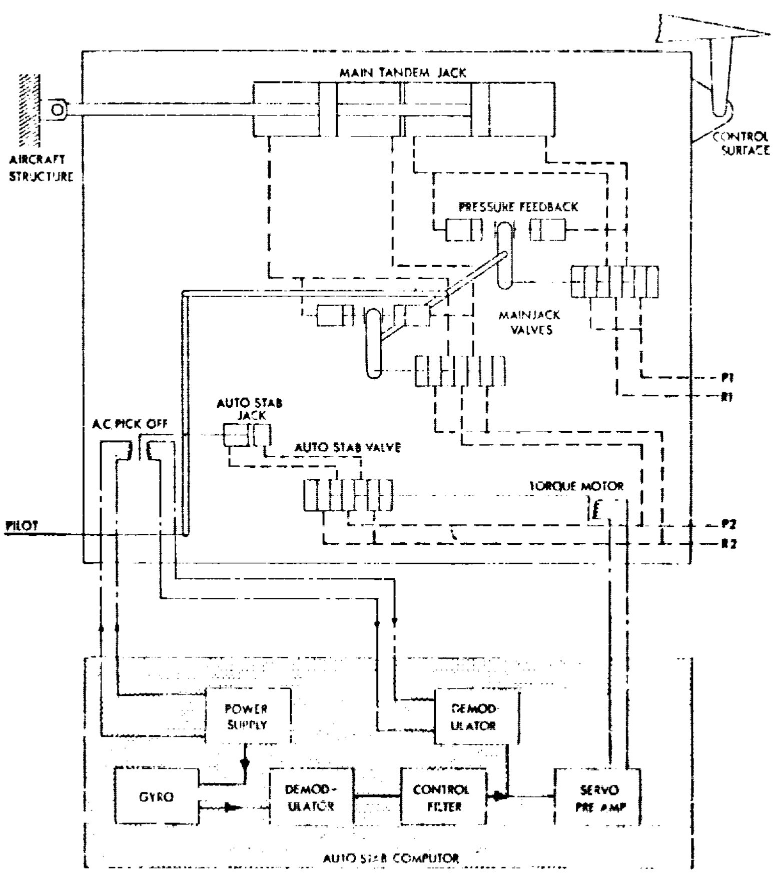


FIGURE A-120 POWER CONTROL UNIT SCHEMATIC DIAGRAM

CONCEPT 78-160 HARRIER REACTION CONTROL SYSTEM

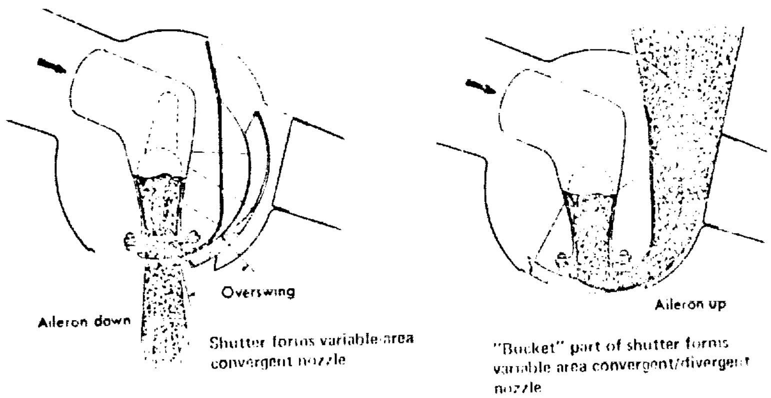
The extraordinary speed range of the Harrier has necessitated the adoption of an unusual flying controls system. In fact, the aircraft uses two different methods of controlling motions about the three axes. There are the conventional aerodynamic moving surfaces for control during fully wing-borne flight and a reaction control system for use during the V/STOL modes. During the transition between fully wing-borne and fully jet-borne flight, the combined power of conventional and reaction controls is, of course, superior to the control powers normally available to a pilot during approach conditions. Air jet reaction control valves situated at the extremities of the aircraft controlled from the normal pitch, roll and yaw circuits are used for V/STOL modes.

Downward-blowing valves at the nose and tail provide pitch control. Sideways-blowing valves at the tail provide yaw control.

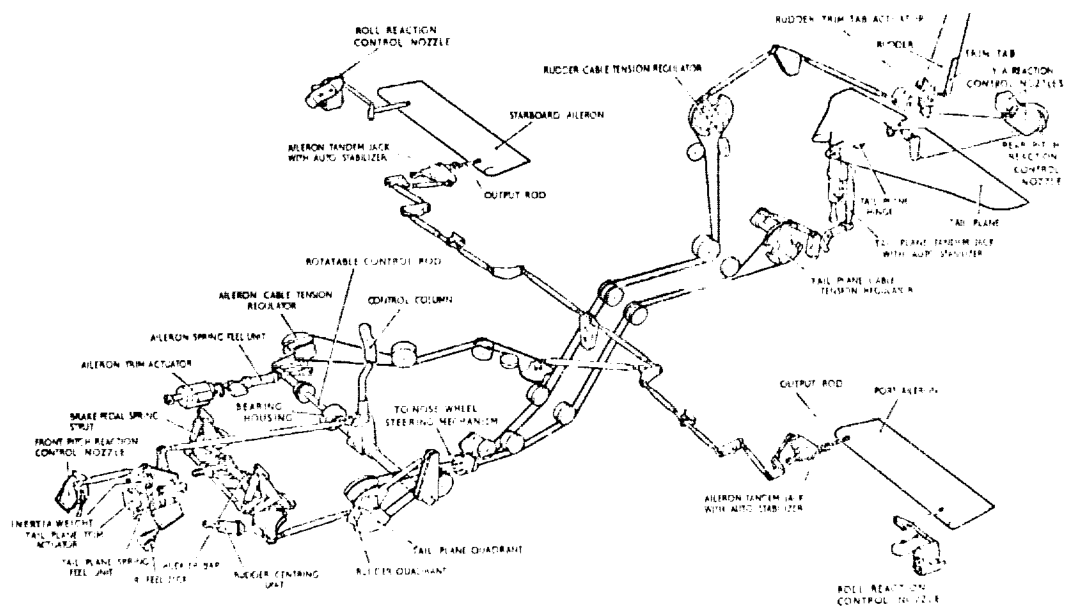
Figure A-121A shows that the valves that blow upwards and downwards at each wing tip to provide roll control.

Figure A-121B shows the interconnection with the appropriate conventional control circuits. It should be noted that the front-pitch and the yaw valves do not connect to a power-operated control surface. Duct routing is shown in Figure A-121C.

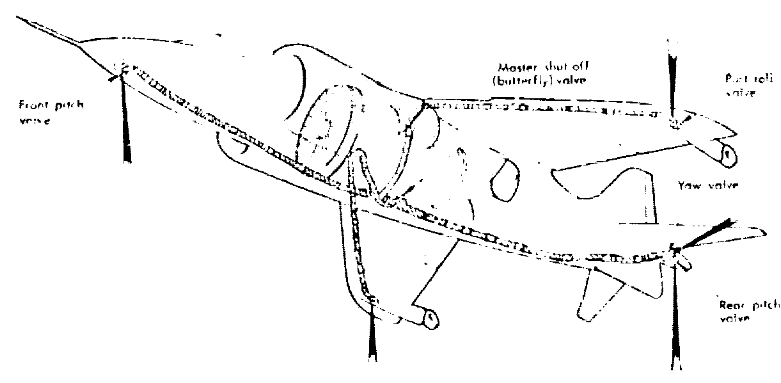
At the maximum reaction control power demand in hover the system is transmitting energy at the rate of about 3,000 h.p. (2,200 Kw.). The installed weight is a little over 200 lbs. (90 kg.).



A. Wing tip roll reaction control valves.



B. The layout of the flying control system showing the conventional and reaction control circuits.



C. Duct routing of the reaction control system.

FIGURE A-121 HARRIER REACTION CONTROL SYSTEM

CONCEPT 79-158 HARRIER NOZZLE ACTUATION SYSTEM

INTRODUCTION

The nozzles are rotated by shafting and chains, powered by an air motor, and controlled by a single lever in the cockpit. The complete system is termed the Engine Nozzle Actuation System, and pilot selection and air motor output comprise the input and error signals of a simple servo loop controlling engine nozzle position.

SYSTEM DESIGN

The Plessey Co. Ltd. were producing air motor units for engine reheat nozzle drive, and Plessey were asked to investigate the possibility of providing a power unit, containing two such air motors. At the same time, the remainder of the drive system was being designed at Kingston. The design philosophy incorporates torque tubing, hooks joints and chain and sprocket drives, see Figure A-122.

PNEUMATIC SYSTEM

Air is bled from the 6th stage of the engine compressor, via flexible piping, to a pressure regulating valve controlling inlet pressure to the air motor at a nominal 35 lb/sq in gauge (2.46 Kg/CM^2). The air motor is air frame mounted, trunnion supported at one end and driving into an auxiliary gear box, also airframe mounted, through a splined drive. The splined drive is a quill shaft, designed to fail at a torque in excess of the nominal maximum stalled torque of the air motor but less than the mechanical transmission system. An air filter is upstream of the regulating valve and a fixed bleed path is incorporated into the valve so that in the event of a failed shut valve, sufficient pressure would exist at the air motor to retain the engine nozzles in the last selected position.

AIR MOTOR SERVO UNIT

The Air Motor Servo Unit is the prime mover in the nozzle actuation system. Operation of the unit is explained with reference to Figure A-123.

Air at controlled pressure enters the unit, through a built-in 40 micron steel mesh filter, to the rotary control valve. With the system at rest, this valve is closed. A positional demand made by the pilot is transmitted through cables and pulleys to a differential gearbox, the second input of which is stationary. The output from this differential opens the rotary control valve, allowing air to pass to duplicated Rootes blowers working in reverse as air motors. The energy in the air is converted to mechanical form, and the exhaust air passes back through the rotary control valve to atmosphere. The drive from the two air motors is summed in an epicyclic gear box; should one air motor or its output shating seize the remaining motor will maintain the output torque, but the final drive out of the gear box will rotate at only half the previous speed. A feed back drive from this output forms the second input to the feed back differential, cancelling the original input as the motor rotates, and thus re-centering the rotary control valve to bring the system to rest. In practice, the response is such that at the designed supply pressure the pilot is unable to beat the system.

PILOT CONTROL MECHANISM

The pilot control consists of a single lever in the cockpit using push rods and cables working in tension for both directions of travel.

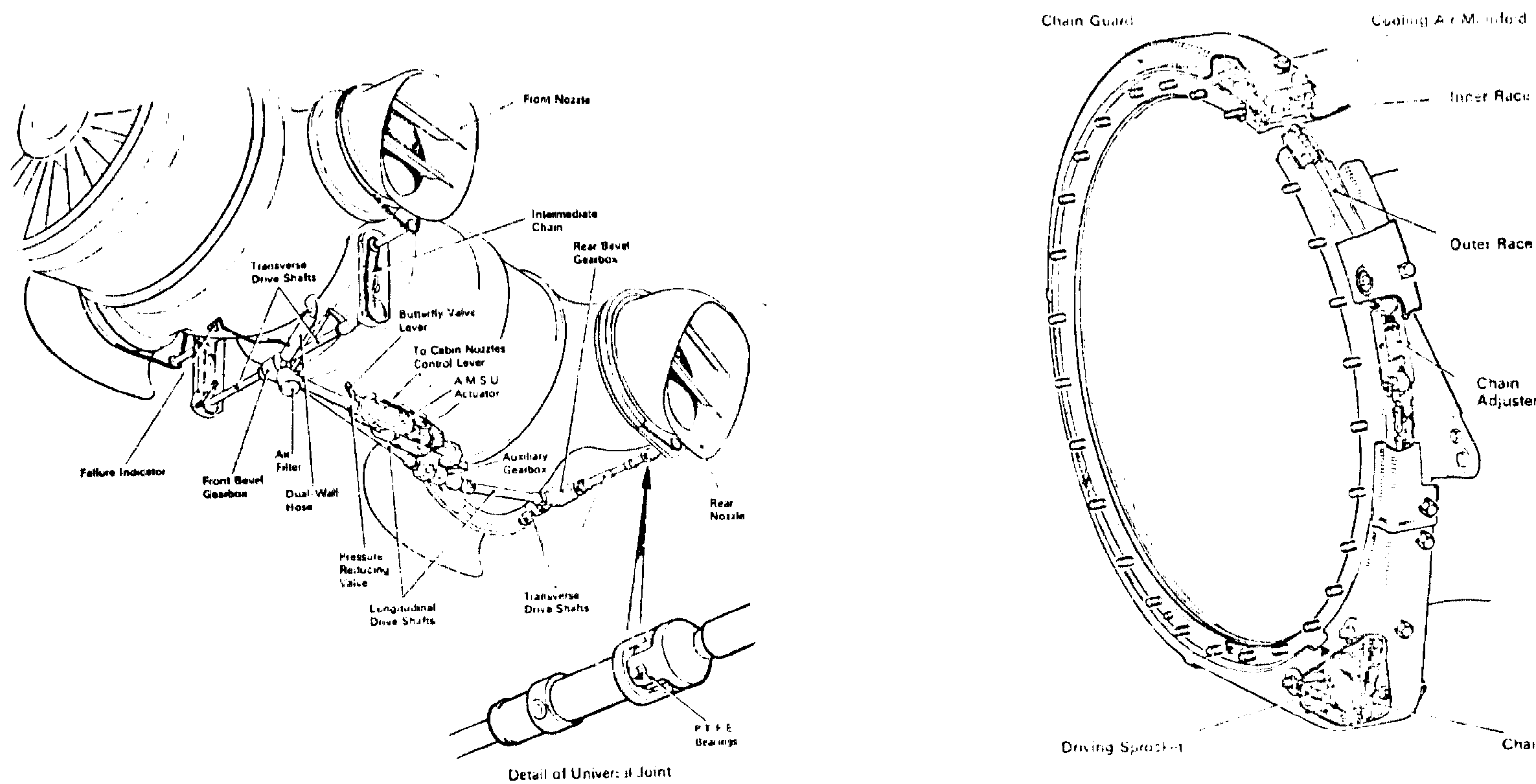


FIGURE A-122 LAYOUT OF THE HARRIER DRIVE SYSTEM AND DETAIL OF THE NOZZLE BEARING.

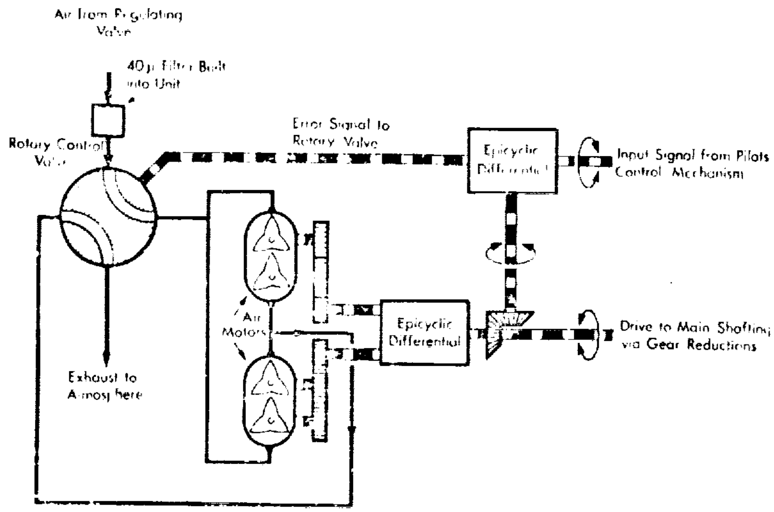


FIGURE A-123 SCHEMATIC DIAGRAM OF THE NOZZLE ACTUATION

CONCEPT 80-026 SIMPLEX PACKAGE 680J

The Simplex package is a surface actuator controlled by mechanical and electrical inputs through a servo valve. The package is normally powered by two conventional central hydraulic systems, but has the additional feature of an integral Emergency Hydraulic System (EHS). In the event of loss of both central hydraulic systems, the EHS is designed to provide a get-home-and-land capability.

The surface actuator portion of the Simplex package, like the F-4 production actuator, consists of dual tandem cylinders. A two-piece stainless steel (rather than one-piece aluminum) body was selected to enhance survivability and reliability. In normal operation one cylinder is powered by the airplane's conventional PC-1 central hydraulic system while the other cylinder is powered by the airplane's conventional PC-2 central hydraulic system. Control and power requirements of the Simplex package are the same as those specified for the F-4 production actuator. The auxiliary RAM, lock-up device, and solenoid valves used by the Automatic Flight Control System (AFCS) and the Stability Augmentation System (SAS) are incorporated into the Simplex package.

The EHS is composed of the following basic components: motor, pump, reservoir, switching valve and two pressure switches, see Figure A-124. Control of the surface actuator during EHS operation is the same as that for the conventional system. Power for the surface actuator to provide get-home-and-land capability is supplied by the motor-pump-reservoir combination.

EHS Operation

The loss of central hydraulic system pressure is sensed by pressure switches located in the PC-1 and PC-2 pressure ports. Figure A-125 contains a hydraulic system diagram of the Simplex package. In operation, when the pressure at either pressure switch drops below a safe level, the switch closes, completing the electrical circuit to the contactor relay to supply electrical power to the EHS motor-pump. If PC-2 pressure is lost, the EHS goes into standby. If PC-1 pressure is lost, a switching valve ports the EHS output into the PC-1 portion of the package. The package was not designed to permit porting the EHS output into the PC-2 portion.

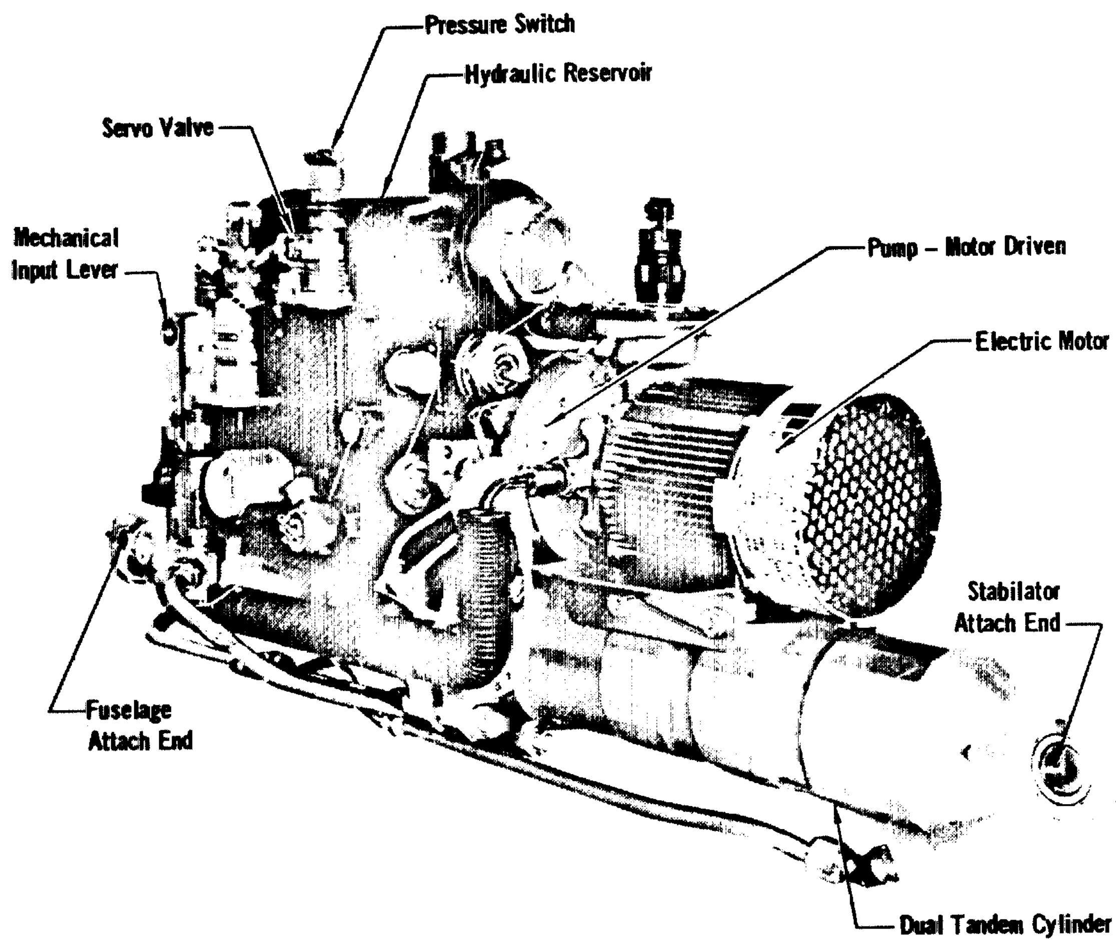


FIGURE A-124 SIMPLEX PACKAGE

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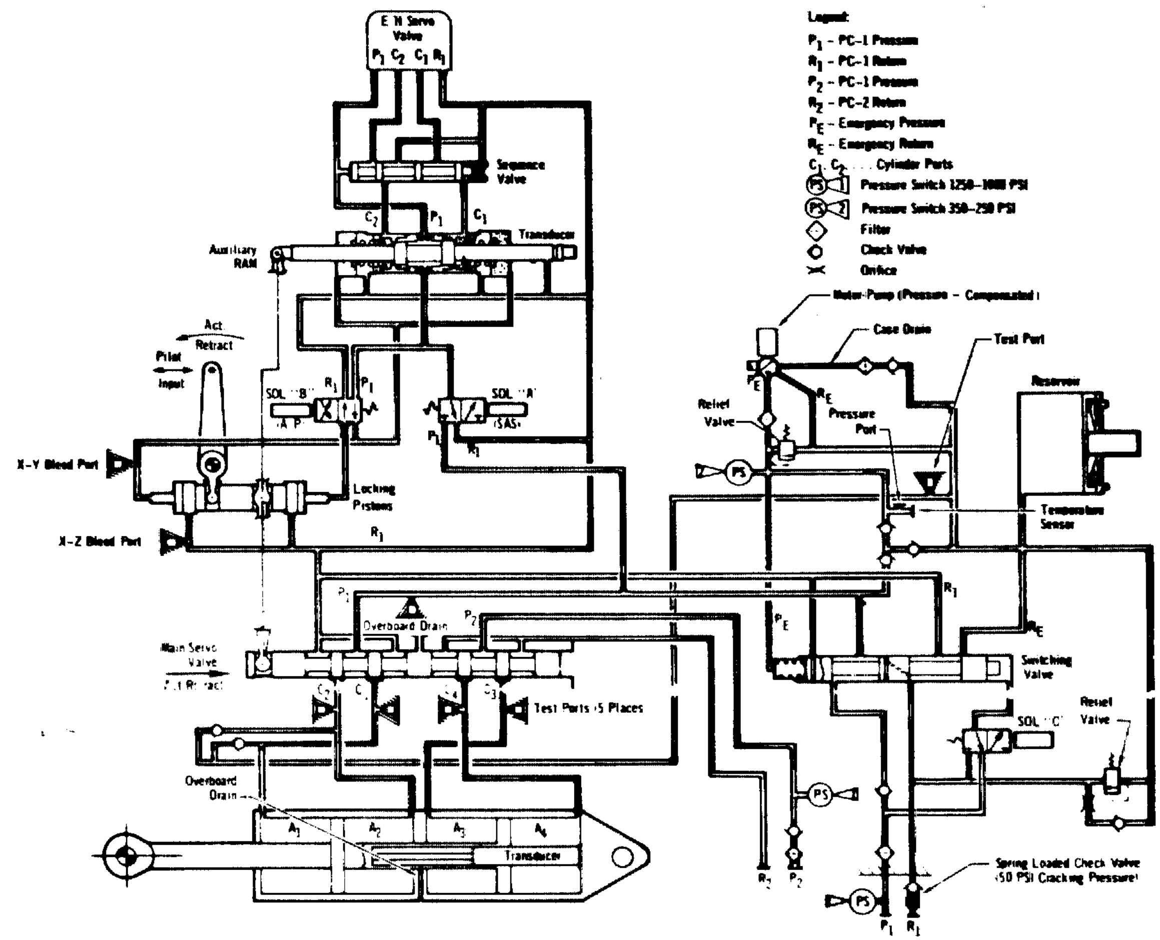

FIGURE A-125 HYDRAULIC DIAGRAM-SIMPLEX PACKAGE

CONCEPT 81-027 INTEGRATED SERVO PUMP ACTUATOR IAP

The integrated servo pump actuator package is a unit which utilizes electrical power and electrical control signal commands to provide an output RAM position to position an aerodynamic control surface.

The input is a position command in terms of an electrical signal. This command is then amplified through a gain term and applied to a summer circuit. The resulting signal, which is electrical, is applied to a servo amplifier or operational amplifier. The resulting output of the servo amplifier is an error signal which in turn commands the servo valve to provide an output flow to the servo pump. The servo pump in turn provides the main output flow to the hydraulic circuit and load valve configuration and then to the actuator RAM where the actuator provides the position and the output load force. The position of the actuator is fed back by use of a linear variable differential transducer (LVDT) and is negatively fed back into the summer circuit. The yoke from the servo pump also contains a LVDT transducer which feeds back the servo pump yoke position in a negative manner to the summing circuit.

When the control pressure signal commands the control piston, the piston forces the yoke to an off-center position, causing the servo pump to provide output flow in one direction. When the control signal is removed, the return spring forces the yoke over-center so that the position of the inlet and outlet ports reverse. Thus, the servo pump unit by virtue of control of the over-center yoke can be made to provide flow in either direction in the hydraulic circuit.

In the actual hardware, a push-push type circuit was applied, that is, two control pistons were used on each side of the yoke so that the differential servo valve output was always applied to the yoke and the return spring force was only used for center balancing.

System Operation

Figure A-126 illustrates the schematic circuit of the integrated servo pump actuator flight control system. The system consists of an aircraft quality 400 Hz electric motor (9), including a cooling blower, an over-center type servo pump (2) with a yoke feedback transducer (8), an auxiliary pump (3), an electro-hydraulic servo valve (5), a load valve (4), an output actuator RAM (10) with load position feedback transducer (7), a low pressure relief valve (13), filters in both the servo pump and auxiliary pump circuits, bypass valve (11), and check valves. In addition, high pressure relief vales (12) were added to protect the system from over-pressurization.

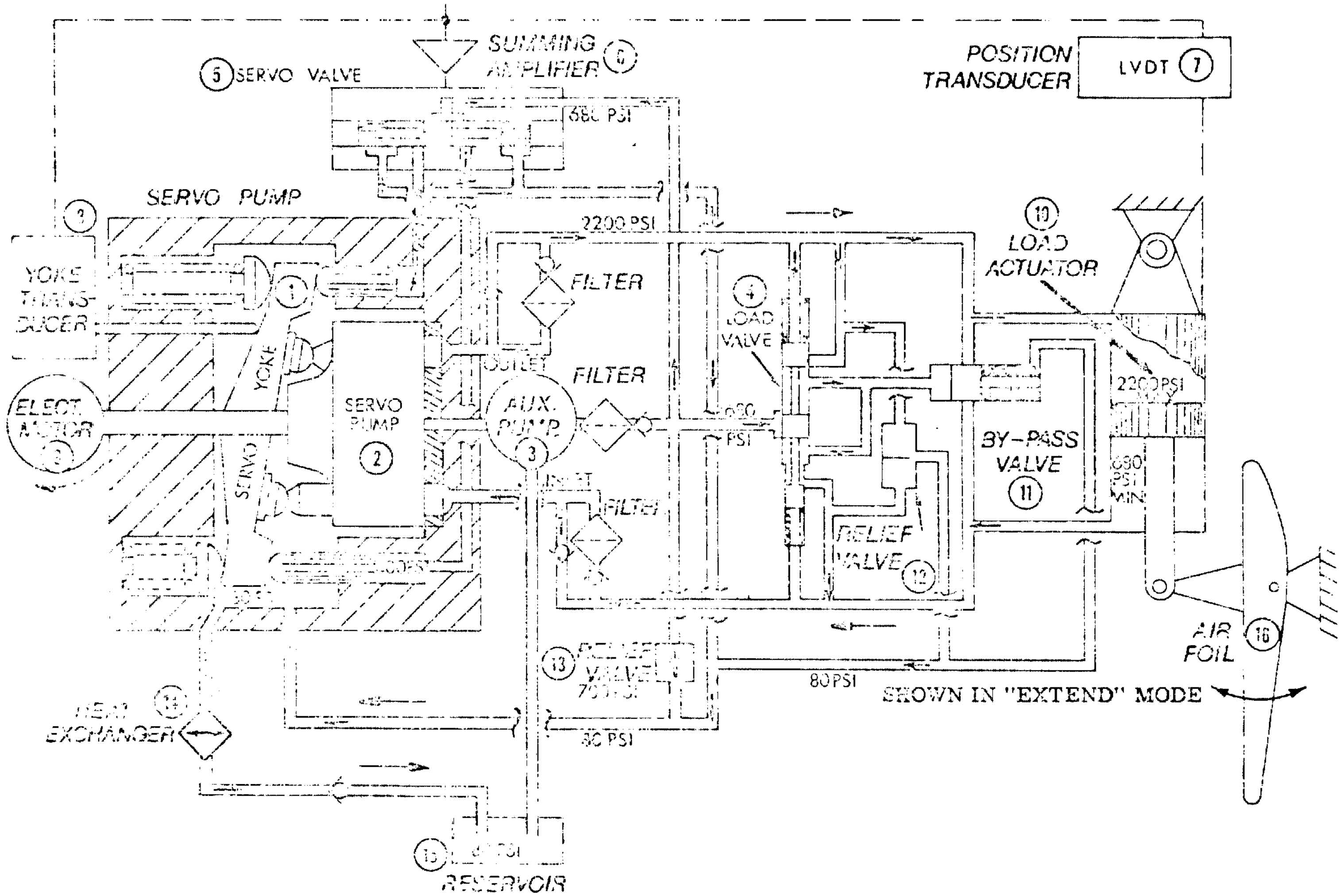

FIGURE A-126 SCHEMATIC CIRCUIT-INTEGRATED SERVO PUMP A/C FLIGHT CONTROL SYSTEM

CONCEPT 82-105   DUPLEX PACKAGE

The General Electric Duplex Package contains two channels, called No. 1 and No. 2. Each channel is independent of the other except in the area of the supply switching valve and the mechanical output from the servo that strokes the pumps. Each channel contains a 400 Hz, 200v, 3Ø motor. Each motor is 10 hp special duty, 11,400 rpm. Mounted directly to the motor is the dual element, radial, ball-piston pump. One element is a constant pressure, variable flow pump whose function is to supply the dual fault servo and the pressurizing reservoir. This pump is rated at 2700 psi. It's capacity of one gpm is ample to run the total servo alone. The second pump element is flow reversible and is rated at 15 gpm.

Integral with the pump housing is a pressurizing reservoir of 25 cubic inch capacity. This reservoir provides pump charging pressure, leakage make-up, a variable container for oil during the stroking of the dissimilar volume power actuator, and thermal expansion provision. The reservoir provides a visual indication of contained oil.

The dual fault correction servo converts the redundant input electrical signals into its output ram motion which drives the flow reversible pumps. The pump stroking mechanism is of the rack and pinion type. It contains shear pins to allow one channel operation in the event the other jams. Backlash is eliminated by a pressure loaded piston pushing against the pump race.

The make-up valve translates to allow the excess oil from the extended cylinder to flow into the reservoir during a retract stroke. During an extend stroke, it allows the pump to draw the added oil it needs from the reservoir.

The force limiting valve is set at an output ram force above any aircraft requirements. A basic function of this valve is to protect the system from high pressures in the event an input signal is accidently introduced that, would cause the ram to bottom. The nominal setting of this valve is 2000 psi.

The by-pass valves cause interconnection of the cylinders of a channel should its servo supply pressure drop below usable limits. Virtually all failures in the motor-pump system will cause a loss of pressure. The nominal setting of this valve is 1800 psi.

The supply switching valve will permit the dual fault correcting servo to operate normally by switching oil supplies. During normal operation, P2 supplies one channel of the servo, while P1 supplies the other two channels. Since a failure of P1 (single failure) would cause two channels to become inoperative, P2 is switched in. Associated with this valve are two check valves used to isolate the pressure systems.

The high-pressure relief valves are set nominally at 3300 psi.

CONCEPT 83-105 TRIPLEX PACKAGE

The assembly of the Triplex Unified Actuator Package (TUAP) is shown in Figure A-127. Figure A-127 shows the essential parts of the package such as the dual power cylinder, motor-pump-reservoir packages, Dual Fault Correcting Auxiliary actuator (DFCA) and switching valves. An explanation of the package is given in the following paragraphs.

There are two primary channels, No. 1 and No. 2, which are normally connected to the dual power cylinder with the No. 3 system operating in a standby condition ready to switch in if a failure occurs. The reversible flow element of the No. 3 system is bypassed during normal operation while the constant pressure (P3) element supplies a third pressure to the auxiliary actuator. Each channel is independent of the others except in the area of the switch valves, intersystem transfer valves and the mechanical output from the DFCA that strokes all three pumps. Each channel contains a 400 Hz, 200 V, 3Ø motor.

Mounted directly to the motor is the dual element, radial, ball-piston pump. One element is a constant pressure, variable flow pump whose function is to supply one channel of the dual fault servo and to pressurize the reservoir. This pump is rated at 2700 psi and 1.0 gpm. The second pump element is flow reversible and is rated at 15 gpm.

Integrated with the pump housing is a pressurized reservoir of 25-cubic inch capacity. This reservoir provides pump charging pressure, leakage makeup and thermal expansion provision. The reservoir provides a visual indication of contained oil.

The dual fault correction servo converts the redundant input electrical signals into its output ram motion which drives the strokable-flow reversible pumps. The pump stroking mechanism is of the cam and follower type. It contains shear pins to allow one channel to operate in the event the other jams. Backlash is eliminated by a pressure loaded piston pushing against the pump race.

The make-up valve is a connected check valve which translates to allow any excess oil from the extend cylinder to flow into the reservoir during a retract stroke. During an extend stroke, it allows the pump to draw any added oil it needs from the reservoir.

The force limiting valve is set at an output ram force above any aircraft requirements. A basic function of this valve is to protect the system from high pressures in the event an output signal is accidently introduced that would cause the ram to bottom. The setting of this valve is 2000 psi.

The by-pass valves cause interconnection of the cylinders of a channel should its servo supply pressure drop below usable limits. This valve will protect against troubles in the motor-pump system since virtually all failures will cause a loss of pressure. The nominal setting of this valve is 1000 psi.

The No. 3 motor-pump package is normally running in a standby mode with the small pump element providing a constant pressure (P3) to one of the three cylinders in the DFCA. The reversible element of the pump is bypassed if both P1 and P2 are present at the P3 switch valve. The bypass valves are not cocked and that EC3 and RC3 are connected to the reservoir pressure R3. If P3 pressure fails, it indicates a failure in the No. 3 motor-pump system that could be caused by a power, motor or pump failure. The primary reasons for using an active standby No. 3 system are:

- The P3 pressure provides a signal for failure monitoring and
- provides a third pressure for the triplex auxiliary actuator.

The switching valve arrangement is set up so that if the No. 1 or No. 2 systems fail with P1 or P2 decreasing to reservoir pressure, then the strokable pump element of the No. 3 system is not bypassed. Now if No. 1 system fails (P1 = R1), then control pressure C1 is blocked off and bypassed with C3 switched into the normal No. 1 side of the cylinder. If P2 then fails (No. 2 system fails second) C2 is bypassed, but C3 is inhibited from switching into the normal No. 2 side of the cylinder. However, if No. 2 system fails first, C2 is bypassed and blocked with C3 allowed to switch into the normal No. 2 cylinder. Now if No. 1 system fails second, C3 will switch into the No. 1 cylinder and then C3 will be taken out of the No. 2 side so that the slew rate of the actuator is not degraded. Using this switch valve arrangement, it can be seen that the No. 3 system will always be available for controlling the output and doesn't depend upon the sequence of normal system failures.

Although the three motor-pump systems are essentially independent and isolated, there is a possibility of some intersystem leakage across lapped fits in the switching valves. Thus intersystem transfer valves are provided so that if one system reservoir becomes overfilled it can transfer oil over into the other systems. These valves are set at a lower pressure than the overboard drain valve so that all systems are filled with oil before any excess oil is dumped overboard.

Filters are included to protect the servovalves in the auxiliary servoactuator. These have 15 micron-absolute elements. The probes of the thermostats are in the return system oil. The return system oil is the initial concentration of heat.

Not shown on the hydraulic schematic, but contained within the package, are two tube-axial, 400 Hz, 200V, 3Ø, 194 cfm cooling fans. These fans blast air over the reservoirs, oil lines and power actuator. A design goal is to avoid the use of separate air to oil heat exchangers. The package is compact which allows the two high-volume fans to direct their air flow over the finned reservoirs, the cylinders of the power actuator and the tubing oil distribution system.

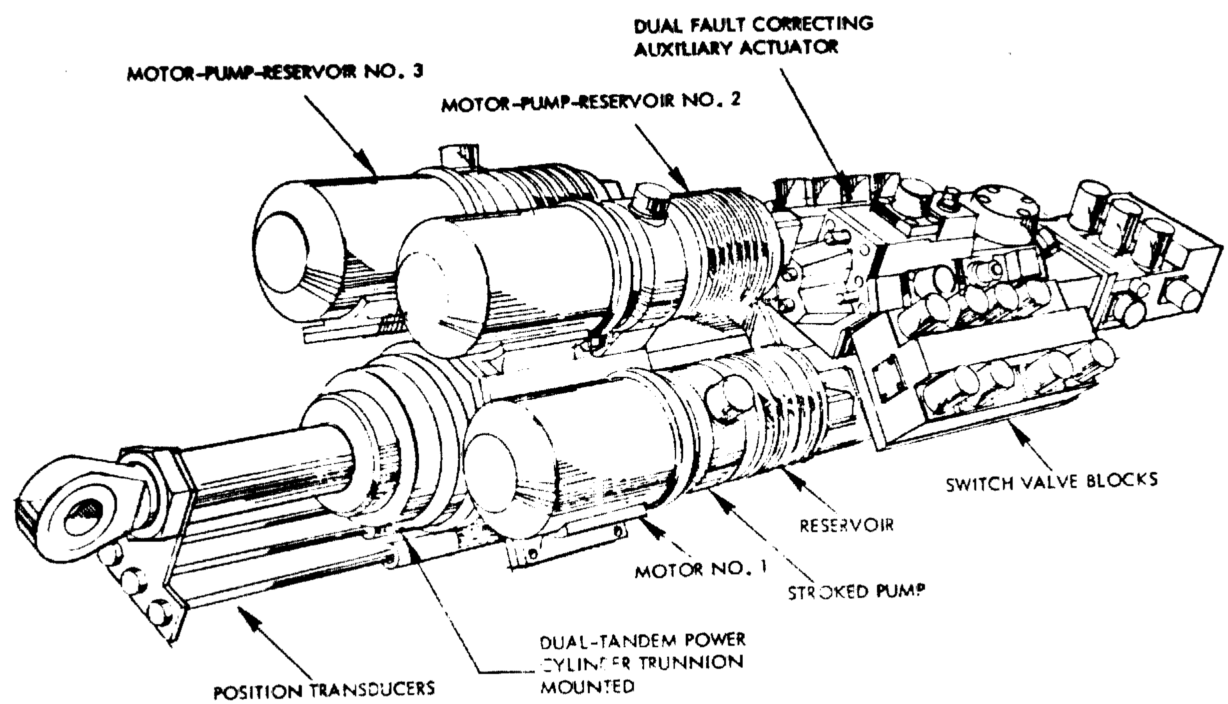

FIGURE A-127 TRIPLEX ACTUATOR PACKAGE

APPENDIX B

LITERATURE SEARCH

BIBLIOGRAPHY

LITERATURE SEARCH DATA FORMAT

3 DIGIT SEQUENCE (FIND) NUMBER

TITLE

ACCESSION NUMBER

1 DIGIT SEARCH CODE (SEE BELOW)

011 DESIGN & EVALUATE SINGLE AXIS REDUNDANT FLY-BY-WIRE SYSTEM R L-099718

VC SETHRE & RV HUPP MCDONNELL DOUGLAS CORP LONG BEACH CALIF

R AFFDL-TR-68-81 1268 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

SPONSORSHIP OR AIRCRAFT ASSOCIATION

YEAR } DATE OF DOCUMENT
MONTH

COMPANY

AUTHOR(S)

DOCUMENT NUMBER OR SOURCE

1 DIGIT DOCUMENT TYPE CODE (SEE BELOW)

SEARCH CODE

A = AIR FORCE FLIGHT DYNAMICS LAB
D = DEFENSE DOCUMENTATION CENTER
M = MECHANIZED INFORMATION CENTER, OSU LIBRARIES
N = NASA
R = ROCKWELL INTERNATIONAL TECHNICAL INFORMATION CENTER

DOCUMENT TYPE CODE

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P = PAPER OR PROPOSAL
R = REPORT
S = STUDY
M = MAGAZINE
L = LETTER
B = BROCHURE
H = HANDBOOK
D = DATA SHEETS
C = CHARTS BRIEFING

000 PRACTICAL CONTROL SYSTEM COMPONENTS N A64-25867
EJ KOMPASS CONTROL ENGINEERING
A CONTROL ENGINEERING V11 0964

001 COMPUTER CONTROLLED DIGITAL SERVO N A65-11347
THADDEUS C ROSS INFRARED INDUSTRIES INC SANTA BARBARA CALIF
P NATIONAL ELECTRONICS CON 1064

002 FLY-BY-WIRE-TRIPLE REDUNDANT A-C SERVO ACTUATOR N A65-10388
ARTHUR H DELMEGE SPERRY RAND CORP DETROIT MICH
A SPERRY ENGRG REVIEW 64

003 HYDRAULIC DIGITAL ACTUATOR N A65-17156
DELMEGE & TREMBLAY VICKERS INC TROY MICH DIV OF SPERRY RAND
A CONTROL ENGINEERING 0265

004 ANALYSIS &SYNTHESIS OF A SEQUENTIAL DIGITAL SERVOMECHANISM N A66-34009
SOLANKI & STEARMAN AIR FORCE TECH COLL INDIA COLL OF AERONAUTICS ENGLAND
P SOCIETY OF INSTR TEC V18 0166 NATIONAL AERONAUTICAL LABORATORY OF INDIA

005 FLY-BY-WIRE TECHNIQUES R L-095545
MILLER & EMFINGER SPERRY PHOENIX COMPANY PHOENIX ARIZ
R AFFDL-TR-67-53 0767 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

006 INVESTIGATION OF ROTARY ACTUATION TECHNIQUE R REF DOC
RG READ BENDIX RESEARCH LABS SOUTHFIELD MICH
R 4093 0767 AIR FORCE AEROPROPULSION LAB WPAFB OHIO

007 A FLUIDIC ACTUATOR FOR DIRECT DIGITAL CONTROL N A67-41710
CK TAFT & EA OWEN CASE INSTITUTE OF TECHNOLOGY
A INSTRUMENTATION TECH V14 1067 ARO CORPORATION BRYAN OHIO

008 INVESTIGATE&DEMO TECHNIQUES PRACTICAL APPLY REDUND FLT CONT R REF DOC
BERQUIST COHEN LAHN HONEYWELL INC MINNEAPOLIS MINN
R AFFDL-TR-67-61 67 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

009 PICKING REDUNDANCY LEVEL OF FLIGHT CONTROL ACTUATORS N A68-26861
FC NEEBE AVIONICS CONT DEPT GENERAL ELECTRIC CO
A SPACE/AERONAUTICS V49 0468

010 DESIGN PRIMER FOR FLIGHT CONTROL ACTUATORS N A68-22826
DM LONGYEAR BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
A HYDRAULICS &PNEUMATICS 0368 F111 AIRCRAFT

011 DESIGN & EVALUATE SINGLE AXIS REDUNDANT FLY-BY-WIRE SYSTEM R L-099718
VC SETHRE & RV HUPP MCDONNELL DOUGLAS CORP LONG BEACH CALIF
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012 DIFF PLM PNEU SERVO FLOATING-FLAPPER-DISC SWITCHING VALVES N A68-33910
GOLDSTEN RICHARDSON FOSTER-MILLER ASSO WALTHAM MASS & MIT CAMBRIDGE MASS
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013 FLUIDIC VORTEX VALVE SERVOACTUATOR DEVELOPMENT R H-023213
HONDA & RALBOVSKY GENERAL ELECTRIC SCHENECTADY NY
R USAAVLABS TR69-23 0569 US ARMY AVIATION MTRL LAB FORT EUSTIS VA

014 TOWARDS DIGITAL CONTROL IN AIRCRAFT N A69-40485
PA HEARNE ELLIOTT FLIGHT AUTOMATION LTD ROCHESTER KENT ENGLAND
P INT AEROSPACE ABSTRACTS 0969

015 FLUIDIC LOGIC FOR A PNEUMATIC STEPPING MOTOR N A69-34310
GRIFFIN & COOLEY NORTHROP CORP LAB & AUTONETICS DIV ROCKWELL INTL
A INSTRUMENTS & CONT SYS 0669 NASA LEWIS RESEARCH CENTER

016 FLY-BY-WIRE REDUNDANT ACTUATOR R REF DOC
JF EMFINGER SPERRY PHOENIX COMPANY PHOENIX ARIZ
A SPERRY ENGRG REVIEW V22 69 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

017 MACHINE DESIGN 1970 FLUID POWER REFERENCE ISSUE R REF DOC
A PENTON PUBLICATION CLEVELAND OHIO
M VOLUME 42 NO 27 0970

018 RESEARCH FLT CONT SYS VOL 3 FLY-BY-WIRE TECHNIQUES R L-104293
BAZILL & JENNY HYDRAULIC RESEARCH & MFG CO VALENCIA CALIF
R AFFDL-TR-69-119 1070 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

019 PRI FLT CONT CKT DEVELOP FROM MECH SYS TO DIGITAL SIGNAL SYS N N70-23030
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020 DIGITAL DATA PROCESSING IN AUTOMATIC FLIGHT CONTROL SYSTEMS N N71-12616
RW HOWARD ELLIOT FLIGHT AUTOMATION LTD ROCHESTER KENT ENGLAND
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021 747 FLIGHT CONTROLS ADVANCE HYDRAULICS STATE-OF-THE-ART N A70-28019
AB BILLET BOEING CO COMMERCIAL AIRPLANE DIV SEATTLE WASH
A SAE JOURNAL VOL 78 0470 BOEING 747 AIRCRAFT

022 PNEUMATIC & HYDRAULIC ACTUATORS R REF DOC
D TAYLOR UNIVERSITY OF ------CLYDE SCOTLAND
A CONTROL & INSTRUMENTA V2 0870

023 INVESTIGATION OF ROTARY ACTUATION TECHNIQUE
VERGE,READ & SIKORA BENDIX RESEARCH LABS SOUTHFIELD MICH
R AFAPL-TR-70-52 0970 AIR FORCE AERO PROPULSION LAB WPAFB OHIO

024 A ROTARY ACTUATOR FOR SPACE MISSIONS R REF DOC
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A TECH ACCOMPLISHMNTS 1970 71 GODDARD SPACE FLT CENTER GREENBELT MD

025 NEW METHOD PERFORM DIGITAL CONT SYS ATTITUDE COMP QUATERNION N A70-19302
BP ICKES LOGICON INC SAN PEDRO CALIF
A AIAA JOURNAL VOL 8 NO 1 0170

026 SURVIVABLE FLT CONT SYS PROGRAM,SIMPLEX ACTUATOR PACKAGE R L-103917
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R AFFDL-TR-70-135 1170 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

027 RESEARCH,FEASIBILITY INTEGRATED SERVO PUMP ACTUATOR PACKAGE R L-106023
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R AFFDL-TR-71-19 0171 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

028 MULTIMODE FLT CONT DEFINITION STUDY PRECISION WEAPON DELIVER R L-105361
RP QUINLIVAN GENERAL ELECTRIC CO
R AFFDL-TR-71-39 0671 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

029 SURVIVABLE FLT CONT SYS REPORT / STUDIES,ANALYSES & APPROACH R A-033719
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030 SURVIVABLE FLT CONT SYS SUPP 1 CONTROL CRITERIA STUDIES R L-105533
KISSLINGER & WENDL MCDONNELL AIRCRAFT CO ST LOUIS MO
R AFFDL-TR-71-20 SUPP 1 0571 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

031 SURVIVABLE FLT CONT SYS SUPP 2 CONTROL LAW DEVELOPMENT STUDY R L-105534
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R AFFDL-TR-71-20 SUPP 2 0571 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

032 SURVIVABLE FLT CONT SYS SUPP 3 HYD POWER & ACTUATION STUDY R L-105535
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033 HIGH TORQUE CMG ROTARY ACTUATOR R REF DOC
GENERAL ELECTRIC CO AIRCRAFT EQUIP DIV BINGHAMTON NY
R ACS 10357 0672 GODDARD SPACE FLT CENTER GREENBELT MD

034 FAILSAFE HYD ACTUATOR FLT CONT FOR JET AIRCRAFT R REF DOC
A HORSTIN BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
A HYD & PNEUMATICS V25 0872 F14 AIRCRAFT

035 UNIVERSAL DIGITAL AUTOPILOT & INTEGRATED AVIONICS SYSTEM N A73-14735
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P WESCON TECH PAPERS 0972

036 SPACECRAFT DIGITAL ATTITUDE CONTROL N A73-14736
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P WESCON TECH PAPERS 0972 COMSAT & INTELSAT

037 SPACECRAFT CONT HRDW USED WITH DIGITAL PROCESSORS N A73-14737
JC RANDALL TRW SYSTEMS GROUP REDONDO BEACH CALIF
P WESCON TECH PAPERS 0972

038 DEMO OF DIGITAL 3 AXIS SPACECRAFT CONTROL N A73-14738
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039 ELECTRO-RAM A NEW ACTUATOR CONCEPT FOR FLY-BY-WIRE R REF DOC
MITCHELL & WARREN LTV ELECTROSYSTEMS INC
A SIGNAL JUNE 1972 0672 ADP680J & F-4 AIRCRAFT

040 PRODUCING THE RIGHT RESPONSE N A72-34144
H COWIN FAIREY HYDRAULICS LTD ENGLAND
A FLIGHT INTERNATIONA V101 0672 BAC-BREGUET JAGUAR AIRCRAFT

041 TECHNIQUE MEASURE IN-PLANE DISPLACMNTS BY HOLOGRAPH INTERFER R H-027088
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R AFFDL-TR-72-5 0272 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

042 F8 CFBW SYS CHARACTERISTICS FOR IRON BIRD SIMULATOR R REF DOC
LOCK & WHEELDON NASA EDWARDS CALIF
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043 RELIABILITY & REDUNDANCY STUDY FOR ELECTRONIC FLT CONT SYS R REF DOC
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044 ELECTROMECHANICAL ACTUATION SYS FOR SPACE SHUTTLE R REF DOC
GARRETT AIRESEARCH MFG CO LA CALIF
R 73-9151 0473 NASA JOHNSON SPACE CENTER HOUSTON TEXAS

045 HI TEMP ROTARY HYD ACTUATOR FOR HI PERFORMANCE AIRCRAFT R REF DOC
RK VAN AUSDAL BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
R 7-3203 1073

046 EFFECTS OF BIAS ERRORS IN REDUNDANT FLT CONT SYS N A73-25783
RF STENGEL CHARLES STARK DRAPER LAB MIT CAMBRIDGE MASS
A JOURNAL OF AIRCRAFT V10 0373 NASA

047 DIGITAL FLIGHT CONTROL TECHNOLOGY ROADMAP R REF DOC
PE BLATT AIR FORCE FLIGHT DYNAMICS LABORATORY WPAFB OHIO
R AFFDL/FGL-TM-73-65 0973 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

048 DESIGN,DEVELOPE LATERAL AXIS INTEGRATED ACTUATOR R REF DOC
BECKER & PENDERSEN VICKERS A-O-M DIV SPERRY RAND CORP
R AFFDL-TR-73-26 0273 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

049 RESEARCH ON DIGITAL FLT CONT SYS ELECTRO-HYD SERVO VALVE R REF DOC
BANFIELD,CAYWOOD & HYDRAULIC RESEARCH & MFG CO BURBANK CALIF
R ASD-TDR-62-558 0263 FLT CONT LAB AERONAUTICAL SYS DIV WPAFB OH

050 BENCH EVALUATION HRM-C SECONDARY ACTUATOR,PROJ SPACE SHUTTLE R REF DOC
ER SHIMO JR SYSTEMS DEVELOPMENT BRANCH GUIDANCE & CONT DIV NASA
R MSC-07483 0672 MANNED SPACECRAFT CENTER HOUSTON TEXAS

051 BENCH EVALUATION HRM-A SECONDARY ACTUATOR,PROJ SPACE SHUTTLE R REF DOC
ER SHIMO JR SYSTEMS DEVELOPMENT BRANCH GUIDANCE & CONT DIV NASA
R MSC-05806 1271 MANNED SPACECRAFT CENTER HOUSTON TEXAS

052 SIMULATION EVALUATION OF FLIGHT COMMAND FUNCTIONS R AD-869164
STEPHAN & CHANDLER HUGHES AIRCRAFT CO CULVER CITY CALIF
R AFFDL-TR-70-4 0370 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

053 RESEARCH,DEFINE,DEMO SOLID STATE SW&MUX SYS FBW FLT CONT SYS R AD-874532
MRAZEK & ROBERTS GENERAL DYNAMICS FORT WORTH TEXAS
R AFFDL-TR-70-80 0670 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

054 DEVELOPMENT TEST OF A ROTARY MECHANICAL ACTUATOR C-W L16566 R REF DOC
NP JAHN ROCKWELL CORP LA DIV LOS ANGLES CALIF
R TFD-71-1269 0472

055 TEST OF A ROTARY MECHANICAL ACTUATOR C-W 175501 R REF DOC
NP JAHN NORTH AMERICAN AVIATION LA DIV LOS ANGLES CALIF
R TFD-71-991 0771

056 GEAR FAILURE TESTS OF A ROTARY ACTUATOR C-W 175501 R REF DOC
NP JAHN NORTH AMERICAN ROCKWELL LA DIV LOS ANGLES CALIF
R TFD-72-940 0872

057 SYNCHRONIZATION OF MULTICHANNEL ELECTROHYDRAULIC ACTUATORS R REF DOC
RL ANDERSON BERTEA CORP IRVINE CALIF
P SOCIETY AUTO ENGRS A-6 0472

058 FLY-BY-WIRE HELICOPTER CONTROL R REF DOC
BOEING VERTOL CO PHILADELPHIA PENN
A DESIGN NEWS 1173 XCH-62 HEAVY LIFT HELICOPTER

059 SYNTHESIS & ANALYSIS OF A FBW FLT CONT SYS F4 AIRCRAFT R REF DOC
KISSLINGER,VETSCH MCDONNELL AIRCRAFT COMPANY ST LOUIS MO
P AIAA NO 72-880 0872 US AIR FORCE

060 DESIGN & FLT EXPERIENCE WITH DIGITAL FBW APOLLO GUIDANCE F8 R REF DOC
DEETS & SZALAI NASA FLIGHT RESEARCH CENTER EDWARDS CALIF
P AIAA NO 72-881 0872 NASA F8 AIRCRAFT

061 A HYBRID DIGITAL HYDRAULIC SERVO R REF DOC
YOUNT & STIGLIC NORTH AMERICAN AVIATION/AUTONETICS LOS ANGLES CALIF
P X5-355/33 0565

062 FLUID POWER & CONT REQMTS FOR SPACE SHUTTLE ORBITING VEHICLE R REF DOC
MCGILLEN,ROSENBAUM SPACE DIV ROCKWELL INTERNATIONAL
P SAE 730986 1073

063 FLT CONT SYS ADVANCES FOR NEAR FUTURE MILITARY AIRCRAFT R REF DOC
PE BLATT FLT CONT DIV AIR FORCE FLT DYNAMICS LAB WPAFB OHIO
P SAE 730987 1073

064 FLUID POWER & CONT REQMTS FOR US ARMY HEAVY LIFT HELICOPTER R REF DOC
TP PEPPLER BOEING VERTOL CO PHILADELPHIA PENN
P SAE 730988 1073 XCH-62 HEAVY LIFT HELICOPTER

065 FLUID POWER NEEDS COMMERCIAL TRANSPORT AIRCRAFT IN SEVENTIES R REF DOC
GF MOORE TRANS WORLD AIRLINES INC KANSAS CITY MISSOURI
P SAE COMM A6 SYMPOSIUM 1073

066 FLUID POWER NEEDS FOR COMMERCIAL TRANSPORTS R REF DOC
OA DARCEY UNITED AIR LINES
P SAE COMM A6 SYMPOSIUM 1073

067 FLUID POWER NEEDS FOR COMMERCIAL TRANSPORT AIRCRAFT R REF DOC
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P SAE COMM A6 SYMPOSIUM 1073

068 UNIQUE FEATURES OF B1 FLIGHT CONTROL SYSTEMS R REF DOC
JE CAMPBELL NORTH AMERICAN ROCKWELL CORP LOS ANGLES CALIF
P AIAA NO 72-872 0872

069 SERVOACTUATOR STUDY FOR SPACE SHUTTLE APPLICATIONS R REF DOC
GD SMITH CONVAIR DIV GENERAL DYNAMICS SAN DIEGO CALIF
R GDC-DCB70-014 0970 NASA MANNED SPACECRAFT CENTER HOUSTON TEXAS

070 POWER-BY-WIRE & FLY-BY-WIRE FLIGHT CONTROL R REF DOC
VICKERS A-O-M DIV SPERRY RAND TROY MICH
A MARKETING NEWS MN 2D-1 0571 F4 AIRCRAFT

071 PROBLEMS,SOLUTIONS DESIGN CONT AUG SYS LONGITUDE UNSTABLE R REF DOC
LR TOMLINSON BOEING COMPANY SEATTLE WASH
P AIAA NO 72-871 0872 SST AIRCRAFT

072 HYDRAULICS FOR V/STOL AIRCRAFT R REF DOC
BECKER,MATHIS VICKERS DIV SPERRY RAND TROY MICH
P VICKERS FLUID PWR CON 22 1072 V/STOL AIRCRAFT

073 DESCRIPTION, DIGITAL SERVO ACTUATORS MISSILE & AIRCRAFT APPL R REF DOC
CADILLAC GAGE CO DETROIT MICH
P CADILLAC GAGE 0761

074 FLIGHT CONTROLS DAMP BIG AIRCRAFT BENDING R REF DOC
JOHANNES,BURRIS THE BOEING CO SEATTLE WASH
A CONTROL ENGINEERING 0967 LAMS PROGRAM WPAFB OHIO

075 FAIL-OPERATIVE STABILITY AUG SYS WITH SELF TEST CAPABILITY R H-015865
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P 650570 SAE 0765 C141 JET TRANSPORT

076 SURVIVABLE CONTROLS GAIN EMPHASIS R REF DOC
ML YAFFEE AIR FORCE FLT DYNAMICS LAB WPAFB OHIO
A AVIATION WEEK & SPACE 0270 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

077 POTENTIAL OF MONITORLESS CONT SYS OF HI REDUNDANCY EFFICINEC R H-015865
EA OHERN AUTONETICS DIV NORTH AMERICAN AVIATION LOS ANGLES CALIF
P 650574 SAE 0765

078 FLIGHT CONTROL R REF DOC
RN MCCOY CONTROL ENGINEERING,DATA SYSTEMS DIV AUTONETICS
A SPACE/AERONAUTICS R&D 0767

079 DEVELOPMENT OF FAILURE-CORRECTING FLIGHT CONTROL SYSTEM R H-015865
FC NEEBE ARMAMENT & CONTROL PRODUCTS,GENERAL ELECTRIC CO
P SAE 650573 0765

080 ANALYSIS OF TOROIDAL ACTUATION SYS FOR AEROSPACE VEHICLES R H-015865
H DREHER LYCOMING/STRATFORD DIV AVCO CORP
P SAE 650580 0765 NAVWEPS

081 ELECTROHYDRAULIC FLOW-CONTROL SERVOVALVES R REF DOC
SERVOVALVE PANEL COMMITTEE A6 AEROSPACE FLUID POWER & CONT TECHNOLOGIES
P SAE ARP 490B 0470 SOCIETY OF AUTOMOTIVE ENGINEERS INC

082 DIGITAL SERVOVALVE R REF DOC
HYDRAULIC RESEARCH & MFG CO VALENCIA CALIF
P HYD RESEARCH & MFG CO 68 NORTHERN ORDNANCE

083 MULTIPLE-CHANNEL ACTUATION SYS FOR BOEING 2707 SST HORZ TAIL N A70-35827
RH HUFLOW BOEING CO COMMERCIAL AIRPLANE GROUP SEATTLE WASH
P AIAA NO 70-915 0770 BOEING 2707 SST

084 CERTIFIED STICK FORCE AUGMENTATION SYS FOR LIGHT STOL AIRCRA N A70-35821
RL DEVINE ET AL HELIO AIRCRAFT CO BEDFORD MASS
P AIAA NO 70-909 0770 HST-550A STOL AIRCRAFT

085 UNIFIED FLIGHT CONTROL ACTUATOR PACKAGE N A69-35640
RC WILD ET AL GENERAL ELECTRIC CO BINGHAMTON NY
P AIAA NO 69-788 0769 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

086 ANALYTIC DESIGN DIGITAL FLT CONT TO REALIZE FLYING QUAL SPEC N A71-31796
RC MONTGOMERY NASA LANGLEY RESEARCH CENTER HAMPTON VA
P AIAA NO 71-955 0871

087 FLT TEST EXPERIENCE IN DIGITAL CONT OF REMOTE PLIOTED VEHICL N A72-40059
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P AIAA NO 72-883 0872

088 DIGITAL SIMULATION FOR NONLINEAR ANALYSIS INTEGRATED ACTUATO N A71-11783
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P INT AEROSPACE ABSTRACTS 0670

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P INT AEROSPACE ABSTRACTS 0968

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N SASLOVE CONVAIR DIV GENERAL DYNAMICS SAN DIEGO CALIF
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091 A PROTOTYPE FLY-BY-WIRE FLIGHT CONTROL SYSTEM D AD-860587
JE EMFINGER SPERRY RAND CORP FLIGHT SYSTEMS DIV PHEONIX ARIZ
R AFFDL-TR-69-9 0869 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

092 RESEARCH,DEVELOPE INTEGRATED SERVO ACTUATOR PACKAGE FIGHTER D AD-867038
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093 DEVELOPMENT OF A DIGITAL HYDRAULIC SERVO R REF DOC
EN YOUNT DATA SYSTEMS DIV AUTONETICS ANAHEIM CALIF
P X6-66/33 0166 NORTH AMERICAN AVIATION INC

094 LINEAR AMPLIFIERS USING CONTINUOUS REDUNDANCY TECHNIQUES R A67-34847
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P INT AEROSPACE ABSTRACTS 0567

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P AIAA NO 67-553 0867

096 COMPREHENSIVE RELIABILITY ANALYSIS OF REDUNDANT SYSTEMS P A67-26828
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A J SPACECRAFT,ROCKETS V4 0567

097 HIGH PERFORMANCE HYDRAULIC SYSTEMS FOR SST AIRCRAFT R A66-13780
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P INT AEROSPACE ABSTRACTS 1065

098 SELF-ADAPTIVE,TRIPLE-REDUNDANT FLT CONT SYS WITH IC LOGIC R A65-24157
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P INT AEROSPACE ABSTRACTS 0564

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101 FEASIBILITY STUDY,PROPULSION PHASE FLUIDIC CONTROL SYSTEM D AD-850061
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102 A NONVARYING-C* CONTROL SCHEME FOR AIRCRAFT D AD-689731
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103 FEASIBILITY STUDY,SENSING AND AMPLIFICATION PHASES OF FLUID D AD-864862
JG RUPERT SYSTEMS AND RESEARCH HONEYWELL INC MINNEAPOLIS MINN
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104 RESEARCH ON FLIGHT CONTROL SYSTEMS VOL I,FLOW DIFFERENCE D AD-867806
GD JENNY HYDRAULIC RESEARCH & MFG CO VALENCIA CALIF
R AFFDL-TR-69-119-VOL-I 0170 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

105 RESEARCH,DEVELOPMENT OF SERVO ACTUATOR PACKAGE FOR TACTICAL D AD-873136
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R AFFDL-TR-69-129 0670 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

106 MULTI-CHANNEL DIGITAL FLY-BY-WIRE ACTUATION SYSTEM R LET SUR
DO BAUER ADVANCED DEVELOPMENT NATIONAL WATER LIFT CO KALAMAZOO M
L DOB/JC 11-21-73 1173 F14,F15,L-1011 AND H-46 AIRCRAFT

107 DIRECT DIGITAL CONTROL VALVES-SYSTEMS-ACCESSORIES R LET SUR
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B DIGICON DEVICES

108 ADVANCED ACTUATOR CONCEPTS FOR AIRCRAFT FLT CONT SYS R LET SUR
RL KLEINMAN VOUGHT AERONAUTICS DALLAS TEXAS
R 2-53110/1R-2952 0971 NAVAL AIR DEVELOPMENT CENTER WARMINSTER PA

109 MULTI-CHANNEL DIGITAL FLY-BY-WIRE ACTUATION SYSTEM R LET SUR
SG PRYOR III LOCKHEED-GEORGIA CO MARIETTA GEORGIA
R LG1US42-2-1 REVISED 1072 C-5A AIRCRAFT

110 ELECTROHYDRAULIC SERVOVALVE AND AMPLIFIER R LET SUR
SAM GRAY PACIFIC CONTROLS INC BURBANK CALIF
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111 B1 HYDRAULIC SYSTEM DYNAMIC ANALYSIS R REF DOC
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R TFD-72-1479 0473 B1 AIRCRAFT

112 ANALOG COMPUTER MECHANIZATION FOR B1 DYNAMIC ANALYSIS R REF DOC
R FRAZZINI NORTH AMERICAN ROCKWELL LOS ANGLES CALIF
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113 B1 FLIGHT CONTROL SUB-SYSTEMS R REF DOC
N KUSKA NORTH AMERICAN ROCKWELL LOS ANGLES CALIF
R TFD-73-19A 0373 B1 AIRCRAFT

114 FEASIBILITY STUDY,ADVANCED FLIGHT CONTROL ACTUATION SYSTEM R REF DOC
COLUMBUS AIRCRAFT DIV NORTH AMERICAN ROCKWELL COLS OHIO
R NR72H-240 0672 NAVAL AIR DEVELOPMENT COMMAND WARMINSTER PA

115 REDUNDANT ACTUATION SUBSYSTEM TECHNOLOGY FOR FLIGHT CONTROLS R REF DOC
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116 SUMMARY REPORT ON REDUNDANT CONTROL TECHNOLOGY R REF DOC
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117 DEMO OF APPLICATION OF MAJORITY LOGIC FOR INC RELIABILITY R REF DOC
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R ASD-TDR-63-275 0763 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

118 DESIGN TECHNIQUES,LAB DEVELOPE,ELECTRICAL PRIMARY FLT CONT R REF DOC
SETHRE ET AL DOUGLAS AIRCRAFT CO LONG BEACH CALIF
R ASD-TDR-62-46 0462 FLT CONT LAB AERO SYS DIV WPAFB OHIO

119 SURVIVABLE FLIGHT CONTROL SYSTEM DEVELOPMENT PROGRAM R REF DOC
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R FDC/ADPO-TM-70-1 0170 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

120 RELIABILITY PREDICTION TECHNIQUES FOR FLIGHT CONTROL SYSTEMS R REF DOC
GOTTFRIED ET AL BOOZ,ALLEN APPLIED RESEARCH INC
R AFFDL-TR-67-20 0767 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

121 RESEARCH,INVESTIGATE REDUNDANCY TECHNIQUES FOR NONELECTRICS R REF DOC
HOGAN ET AL GENERAL ELECTRIC COMPANY JOHNSON CITY NY
R AFFDL-TR-65-80 0865 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

122 HANDBOOK FOR SYSTEMS APPLICATION OF REDUNDANCY P H-020781
L LAMPERT US NAVAL APPLIED SCIENCE LAB
H REVIEW DRAFT 0866 ASW SYSTEM PROJECT OFFICE

123 MOOG TECHNICAL BULLETINS R REF DOC
MOOG INC CONTROLS DIV EAST AURORA NY
B TECHNICAL BULLETINS

124 REDUNDANT ELECTROHYDRAULIC SERVOACTUATORS R REF DOC
MOOG INC CONTROLS DIV EAST AURORA NY
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125 ENVIRONMENTAL CONT & HYD SYS FAULT DETECT-FAILURE ISOLATION R REF DOC
GARRET AIRESEARCH MFG CO LOS ANGLES CALIF
R 71-7900 1071 NASA MANNED SPACECRAFT CENTER HOUSTON TEXAS

126 ELECTROMECHANICAL ACTUATION SYSTEMS CAPABILITIES SUMMARY R REF DOC
GARRET AIRESEARCH MFG CO TORRANCE CALIF
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127 SPACE SHUTTLE AEROFLIGHT CONTROL STUDIES R REF DOC
F HERCULES MCDONNELL DOUGLAS ASTRONAUTICS COMPANY-EAST
R MDC E0827 0673 ROCKWELL INTERNATIONAL

128 SPACE SHUTTLE AEROFLIGHT CONTROL STUDIES TEST DATA R REF DOC
JA HEDGES MCDONNELL DOUGLAS ASTRONAUTICS COMPANY-EAST
R MDC E0827 SUPP 1 0673 ROCKWELL INTERNATIONAL

129 ROTARY AND LINEAR ENCODER DATA SHEETS R LET SUR
JOHN ROY TELEDYNE GURLEY TROY NEW YORK
L 1173 APOLLO AND OTHER NASA PROGRAMS

130 MULTICHANNEL DIGITAL FLY-BY-WIRE HARDWARE CAPABILITIES R LET SUR
ED BERLLY SINGER AEROSPACE & MARINE SYS GROUP CLIFTON NJ
B VARIOUS 1173

131 HYDRAULIC RESEARCH REDUNDANT SERVOCONTROLS IN SUMMARY R REF DOC
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B 0972 SPACE SHUTTLE,F4,F8,& B47 AIRCRAFT

132 SPACE SHUTTLE SERVOACTUATOR STUDY AN ANNOTATED BIBLIOGRAPHY R REF DOC
N SASLOVE CONVAIR DIV GENERAL DYNAMICS SAN DIEGO CALIF
R GDC-DCB70-013 0770 NASA MANNED SPACECRAFT CENTER HOUSTON TEXAS

133 SPACE SHUTTLE BRIEFING CHARTS EM VS HYD NASA-JSC R REF DOC
JSC NASA HOUSTON TEXAS
C FIRST JSC STUDY NASA MANNED SPACECRAFT CENTER HOUSTON TEXAS

134 EVALUATION OF A HIGH RESPONSE ELECTROHYDRAULIC DIGITAL CONT R N73-20523
RL ANDERSON BERTEA CORP IRVINE CALIF
R 224200-7 0273 NASA MARSHALL SPACE FLT CNTR HUNTSVILLE ALA

135 DEVELOPE SPACE QUALIFIED HI RELIABLE ROTARY ACTUATOR VOL 2 R N72-31499
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R BRL 6048 0672 NASA GODDARD SPACE FLT CNTR GREENBELT MD

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AIR FORCE FLIGHT DYNAMICS LABORATORY WPAFB DAYTON OHIO
R AFFDL-TR-73-41 0473 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

137 DIGITAL FLT CONT SYS FOR HELICOPTER AND VTOL APPLICATIONS N A68-33486
LA KAUFMAN TELEDYNE SYSTEMS COMPANY HAWTHORNE CALIF
P AIAA 0668 US MARINE CORPS IHAS HELICOPTER

138 CURRENT PNEUMATIC & FLUIDIC DEVELOPMENTS,AEROSPACE & FUTURE N A71-36208
BW ANDERSON BRIGHAM YOUNG UNIVERSITY PROVO UTAH
P AIAA 0471

139 DIGITAL CONTROL VALVES N A70-23684
HF FRIEDLAND ET AL PROCESS SYS INSTRUMENTATION INC LAS VEGAS NEVADA
P ISA TRANSACTIONS V8 NO 4 0869

140 CONTROL SYSTEM REDUNDANCY MECHANIZATION STUDY R REF DOC
E TESCH ET AL MINNEAPOLIS-HONEYWELL REGULATOR CO MINNEAPOLIS MINN
R ASD-TDR-62-772 0263 FLIGHT CONTROL LAB AERONAUTICAL SYS WPAFB O

141 ELECTROHYDRAULIC LINEAR STEPPER ACTUATOR ELSA R REF DOC
DBA AIR EQUIPMENT
B MODEL AF 34 502

142 RESEARCH & DEVELOPE SURVIVABLE FLT CONT SYS SECONDARY ACTUAT R REF DOC
JE RINDE GENERAL ELECTRIC COMPANY BINGHAMTON NY
R ACS 10,431 0673 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

143 RESEARCH & DEMO DIGITAL ELECTROHYDRAULIC SERVO CONTROL VALVE R AD-601597
DS SEIDEL HYDRAULIC RESEARCH & MFG CO BURBANK CALIF
R RTD-TDR-63-4240 0564 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

144 DEVELOPMENT OF HI RELIABLE ROTARY ACTUATOR FOR SPACEFLIGHT R REF DOC
PG READ ET AL BENDIX RESEARCH LABS BENDIX CENTER SOUTHFIELD MICH
R BRL 6048 0672 GODDARD SPACE FLT CENTER GREENBELT MD

145 TESTING OF A PNEUMATIC SERVOMECHANISM R REF DOC
VERGE,READ BENDIX RESEARCH LABS BENDIX CENTER SOUTHFIELD MICH
R AFFDL-TR-71-146 0272 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

146 THE DYNAVECTOR DRIVE R REF DOC
BENDIX RESEARCH LABS BENDIX CENTER SOUTHFIELD MICH
R BRL NO 70-2 0673 BENDIX CORPORATION

147 PROCEEDINGS OF THE FLY-BY-WIRE FLIGHT CONTROL CONFERENCE R AD 857288
JP SUTHERLAND AIR FORCE FLIGHT DYNAMICS LAB WPAFB DAYTON OHIO
R AFFDL-TR-69-58 0669 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

148 INVESTIGATE & DEVELOPE REDUNDANT TECHNIQUES DUAL FAULT CORRE R REF DOC
JE RINDE ET AL GENERAL ELECTRIC COMPANY BINGHAMTON NY
R AFFDL-TR-67-17 0567 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

149 FLY-BY-WIRE REDUNDANT MODULAR CONCEPT R REF DOC
BENDIX CORP,ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
B BENDIX CORP NORTH HOLLYWOOD CALIF

150 THE BALANCED POWER ACTUATION CONCEPT N A66-10670
GR TOOMEY NORTH AMERICAN AVIATION INC LOS ANGLES DIV,LA CALIF
P 650579 AIAA B70 AIRCRAFT

151 LIQUID METAL SERVOACTUATION PACKAGES FOR FLIGHT CONTROLS N A70-40785
KUMPITSCH,STURGES GENERAL ELECTRIC CO AVIONIC CONTROLS DEPT
P 700343 AIAA AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

152 FLIGHT CONTROL ACTUATORS FOR MISSILE APPLICATIONS N A68-33428
HALL,SCHREIBER CURTISS-WRIGHT CORP CALDWELL,NEW JERSEY
P AIAA CURTISS-WRIGHT CORP

153 REDUNDANCY MANAGEMENT ELECTRO-HYD SERVOACT BY MATH MODEL REF R REF DOC
RA CAMPBELL ASTRIONICS LABORATORY
R TMX-64625 1171 MARSHALL SPACE FLIGHT CENTER ALABAMA

154 PRELIM TEST OF E/M SERVOACTUATOR SYS PROPOSED FOR SPACE SHUT R REF DOC
GC STREUDING LOCKHEED ELECTRONICS CO HOUSTON TEXAS
R REF MEMO GC-4449-016 0973 NASA SPACE SHUTTLE

155 BALLSCREW LINEAR ACTUATOR R REF DOC
TO PAINE NASA
P PATENT 3,660,704 0572 NASA SPACECRAFT

156 MECHANICAL ACTUATOR R REF DOC
JC JOHNSON JR ET AL NASA
P PATENT 3,460,397 0869 NASA

157 DIGITAL CONTROL OF ANALOG SERVO ACTUATORS R REF DOC
DW LANDAU ROCKWELL INTERNATIONAL PROPULSION & FLIGHT CONTROL
L IL73-243-120-DWL-1 1273 ROCKWELL INTERNATIONAL

158 HAWKER SIDDELEY HARRIER GR MK 1 R REF DOC
WHF ALLAN ET AL HAWKER SIDDELEY AVIATION LTD SURREY ENGLAND
A AIRCRAFT ENGINEERING C12 1269 ROYAL AIR FORCE

159 STUDY OF ELECTRO-MECHANICAL FLT CONT SYS FOR ORBITER WT RED R REF DOC
J WILKOWSKI NORTH AMERICAN ROCKWELL SPACE DIV
R SD-73-SH-0107A 0873 SPACE SHUTTLE ORBITER

160 HARRIER HYDRAULIC SYSTEM AND FLYING CONTROLS R REF DOC
WHF ALLAN ET AL HAWKER SIDDELEY AVIATION LTD SURREY ENGLAND
A AIRCRAFT ENGINEERING C14 1269 ROYAL AIR FORCE

161 FUJITSU PULSE MOTORS TECHNICAL MANUAL R REF DOC
ICON CORP CAMBRIDGE MASS
H USM CORPORATION

162 APPLICATION OF VERY HIGH PRESSURE HYDRAULIC SYS TO AIRCRAFT R REF DOC
DEMARCHI,HANING ROCKWELL INTERNATIONAL CAD COLUMBUS OHIO
R NR72H-20 0372 NAVAL AIR SYSTEMS COMMAND

163 ADVANCED FIGHTER TECHNOLOGY ENTRY SET R REF DOC
C MARTIN AVIATION WEEK & SPACE TECHNOLOGY
A AVIATION WEEK 0873 MCDONNELL DOUGLAS AFTI AIRCRAFT

164 USAF PLANS ADVANCED FIGHTER PROGRAM R REF DOC
B MILLER AVIATION WEEK & SPACE TECHNOLOGY
A AVIATION WEEK 0773 USAF AFTI AIRCRAFT

165 DIGITAL HYDRAULIC VALVING SYSTEM FINAL REPORT R REF DOC
HLM INCORPORATED WALDWICK NEW JERSEY
R REF NAS 8-28166 0873 NASA/MSFC

166 DIGITAL HYDRAULIC VALVING SYSTEM TRADE-OFF STUDY R REF DOC
HLM INCORPORATED WALDWICK NEW JERSEY
R REF NAS 8-28166 0972 NASA/MSFC

167 DESIGN CRITERIA FOR HIGH-AUTHORITY CLOSED LOOP PRI FLT CONT R REF DOC
HENDRICK ET AL HONEYWELL INC MINNEAPOLIS MINN
R AFFDL-TR-71-78 0872 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

168 PAPERS FROM THE JOINT AUTOMATIC CONTROL CONFERENCE OF AACC R REF DOC
AMERICAN AUTOMATIC CONTROL COUNCIL AACC
P IEEE NO 73CHO750-CCSS 0673 OHIO STATE UNIVERSITY COLUMBUS OHIO

169 BACKGROUND INFORMATION AND USER GUIDE FOR MIL-F-8785B ASG R REF DOC
CHALK ET AL CORNELL AERONAUTICAL LAB INC BUFFALO NEW YORK
R AFFDL-TR-69-72 0869 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

170 ROTARY SERVO HYDRAULIC ACTUATOR R REF DOC
MARTIN COMPANY DENVER COLORADO
R TO 9H2-5-169-3 MISSILE APPLICATION

171 ELECTRONIC BREADBOARD FOR ACTUATOR/DATA BUS INTERFACE R REF DOC
MARTIN MARIETTA CORP DENVER COLORADO
P P-70-122 1170 NASA MARSHALL SPACE FLIGHT CENTER

172 BACKUP COMMAND AUGMENTATION SYS FOR F8 DIGITAL FBW AIRCRAFT R REF DOC
SPERRY FLIGHT SYSTEMS DIVISION PHOENIX ARIZONA
R 71-0399-00-00 1073 NASA EDWARDS AIR FORCE BASE

173 DIGITAL ACTUATOR IDEA
LW TAYLOR NASA LANGLEY RESEARCH CENTER HAMPTON VIRGINIA
D SKETCH 1271

174 CONCEPT OF A DUAL CHANNEL CROSS COUPLED CONT SYS QUADRISAFE R REF DOC
DW LANDAU ROCKWELL INTERNATIONAL AUTONETICS DIV ANAHEIM CALIF
L 73-245-040-DWL-15 1073 REMOTELY PILOTED VEHICLE

175 FLIGHT CONTROL TECHNOLOGY FOR M-X,A LOO TO THE FUTURE R REF DOC
ROCKWELL INTERNATIONAL AUTONETICS DIV ANAHEIM CALIF
C BR73-9476201 0973

176 PRECISE POWER CONTROL---ALL MECHANICAL POWER FLOW R REF DOC
HARNED ET AL GENERAL MOTORS CORP WARREN MICH
P JOINT AUTO CONT CON 1963 0663

177 FLIGHT CONTROL SYS ADVANCES FOR NEAR-FUTURE MIL AIRCRAFT R REF DOC
PAUL BLATT AIR FORCE FLIGHT DYNAMICS LAB WPAFB DAYTON OHIO
P SAE 730987 1073 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

178 SELF-CONTAINED ELECTRONIC FLIGHT CONTROL SYSTEM REPORT NO 4 R REF DOC
RABINOWITZ,PATTI KAMAN AIRCRAFT CORPORATION
R G-166 1061

179 ADVANCED FLIGHT CONTROL POWER SYSTEM REDUNDANCY STUDIES R REF DOC
RAMSEY,WAGNER BOEING-WICHITA
R D3-8194 0370

180 AIRCRAFT FLIGHT CONTROL SYSTEMS FIELD SAFETY EXPERIENCE R REF DOC
BIELKA,KAVALOK,REEL AIR FORCE FLIGHT DYNAMICS LAB WPAFB DAYTON OHIO
R AFFDL-TR-72-33 0572 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

181 A&D AND D&A CONVERTERS BRIDGING ANALOG WORLD TO COMPUTER R REF DOC
R ALLEN ELECTRONIC DESIGN NEWS ASSOCIATE EDITOR
A EDN MAGAZINE 0273

182 8-12 BIT SUCCESSIVE APPROXIMATION A&D CONVERTERS R REF DOC
PRECISION MONOLITHICS INC SANTA CLARA CALIF
D MONO AD-124 1073

183 CONTROL-BY-WIRE ACTUATOR MODEL DEVELOPMENT FOR AFCAS R REF DOC
R HANING ET AL ROCKWELL INTERNATIONAL CAD OHIO
R NR73H-107 0174 NAVAL AIR DEVELOPMENT CENTER

184 SYNCHRONIZATION OF MULTICHANNEL ELECTROHYDRAULIC ACTUATORS R REF DOC
RL ANDERSON BERTEA CORP IRVINE CALIF
P SAE A6 COMMITTEE 0472

185 FLY-BY-WIRE SIGNAL CONVERSION TRADE-OFF STUDY R H-028883
SR FOSTER LTV ELECTROSYSTEMS INC GARLAND DIV DALLAS TEXAS
R 416-16108 1168 AFSC INTEGRATED DUPLEX ACTUATOR

186 NEW SERVOVALVES FOR REDUNDANT ELECTROHYDRAULIC CONTROL R H-028880
GARNJOST,THAYER MOOG SERVOCONTROLS INC EAST AURORA NEW YORK
R TECHNICAL BULLETIN 105 0963

187 AUTOMATIC FLT MANAGEMENT OF FUTURE HIGH PERFORMANCE AIRCRAFT N N70-16973
CM PETRIE BOEING CO SEATTLE WASHINGTON
P AGARD ADVISORY REPORT 19 0569 ADVISORY GP FOR AERO RESEARCH AND DEVEL

188 CES DEV FLT TEST MULTIPLEX HARDWARE FOR FLY-BY-WIRE FLT CCNT R REF DOC
MRAZEK / ROBERTS   CONVAIR DIV GENERAL DYNAMICS SAN DIEGO CALIF
R AFFDL-TR-73-43   0473 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

189 FLY-BY-WIRE FLT CONT SYS STUDIES ACTUATCR INTERFACE   R LET SUR
RG GADBOIS   LEAR SIEGLER ASTRONICS DIV SANTA MONICA CALIF
R ASD73-028   0473

190 B-47 FLY-BY-WIRE INVESTIGATION   R REF DOC
GD JENNY   HYDRAULIC RESEARCH / MFG CO VALENCIA CALIF
P SAE A6 COMMITTEE   0470

191 FEASIBILITY STUDY OF AN ALL-FLUID DIGITAL CONT SYSTEM   R REF CCC
VICKERS DIV SPERRY RAND CCRP TRCY MICHIGAN
S REF NAS 14729   0865 NASA LANGLEY RESEARCH CENTER

192 THE SST FLIGHT CONTROL SYSTEM CONCEPT   R A67-35966
JR HORSNELL   THE BOEING CO SEATTLE WASHINGTON
P AIAA NO 67-570   0867

193 SST FLIGHT CONTROLS-CONCEPT AND DESIGN   R R-025163
FS HOLMAN   THE BOEING CO SEATTLE WASHINGTON
P ASME   0668

194 PROGRAM SUMMARY REPORT OF ELECTRO-PNEUMATIC ACTUATION SYSTEM R REF DOC
MC TRAUTNER   PARKER AIRCRAFT LOS ANGELOS CALIF
R EDR5640022   1267 NASA-MSFC

195 AIRCRAFT FLIGHT CONTROL SYSTEM FIELD SAFETY EXPERIENCE   R L-107717
BIELKA KAVALOK /   THE BOEING CO SEATTLE WASHINGTON
R AFFDL-TR-72-33   0572 AIR FORCE FLIGHT DYNAMICS LAB WPAFB OHIO

196 INVEST DEVEL NEW CONCEPT FOR IMPROVE AIRCRAFT ELECT PWR SYS R L-101272
CH LEE   AIRESEARCH LOS ANGELES CALIF
R 68-4176   0569 NASA ELECT RES CENTER CAMBRIDGE MASS

197 DEVELOPMENT OF A 7 BIT PARALLEL BINARY DIGITAL SERVC VALVE   R AD 649944
HYDRAULIC RESEARCH AND MFG CO BURBANK CALIF
R NOTS TP 4298   0367 US NOTS CHINA LAKE CALIF

198 FLT EVAL OF OPTIMAL COMMANDS FOR F-106 MIN TIME CLIMB/ACCEL   R AD 760571
JA SCHIRO ET AL   HUGHES AIRCRAFT CO CULVER CITY CALIF
R AFFDL-TR-73-32   0373 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

199 FAULT-TOLERANT DIGITAL AIRBORNE DATA SYSTEM   R AD 756485
ANDERSON BIRMINGHAM HUGHES AIRCRAFT CO CULVER CITY CALIF
R AFFDL-TR-72-69   0872 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

200 APPROACH TO APPLICATION OF REDUND TECHNIQUES TO FLT CONT SYS R REF DOC
EA O HERN   AUTONETICS DIV ROCKWELL INT ANAHEIM CALIF
P AIAA&ASD   1163

201 TRIPLEX FLY-BY-WIRE HYDRAULIC CONTROL SERVO   R REF DOC
BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
P DC-931   0969

202 FLY-BY-WIRE HYD SERVO ACTUATORS FOR STABILIZER,RUDDER / FLAP R REF DOC
BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
P DC-911   0469

203 SPACE SHUTTLE BOOSTER PITCH&ROLL CONT SURFACE ACTUAT SYS-ETC R REF DOC
BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
R 7-3103   0172

204 STUDY / TEST PROG PROPOSAL FOR REDUND FLT CONT ACTUATOR SYS   R REF DOC
BENDIX ELECTRODYNAMICS DIV NORTH HOLLYWOOD CALIF
P DC-1062   0571 MARSHALL SPACE CENTER HUNTSVILLE ALA

205 A SELF-REORGANIZING DIGITAL FLT CONT SYS FOR AIRCRAFT   R REF DOC
MONTGOMERY/CAGLAYAN NASA LANGLEY RESEARCH CENTER
P AIAA 74-21   0174

206 CONCORDE TAKES TO THE AIR WITH PSEUDO FLY-BY-WIRE CONTROLS P REF DOC
B DARRIEUS SUD AVIATION CORP
P SAE A-18 0867

207 SST LONGITUDINAL CONTROL SYSTEM DESIGN AND DESIGN PROCESSES R AD 9 3365
LR TOMLINSON BOEING AIRCRAFT CO SEATTLE WASHINGTON
R D6-60285 0673 FEDERAL AVIATION ADMINISTRATION

208 DIGITAL FLIGHT CONTROL RESEARCH R H-029155
POTTER STERN ET AL MASS INSTITUE OF TECHNOLOGY CAMBRIDGE MASS
R RE 83 0672 NASA

209 CEV / EVALUATION OF THE TWEAD II FLT CONT AUGMENTATION SYS R AD 905264
CARLETON ET AL AIR FORCE FLT TEST CENTER EDWARDS AFB CALIF
R AFFTC-TD-72-1 1172

210 DESIGN CRITERIA FOR HIGH-AUTH CLOSED-LOOP PRI FLT CONT SYS R AD 903862
HENDRICK ET AL HONEYWELL INC MINNEAPOLIS MINNESOTA
R AFFDL-TR-71-78 0872 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

211 DIGITAL HYDRAULIC SERVOVALVE EVALUATION R AD 861059
KOVALENKO / WENZEL NAVAL WEAPONS CENTER CHINA LAKE CALIF
R NWC-TP-4777 0869

212 CLOSED-LOOP EVAL OF A PLUS OR MINUS 7 DIGITAL SERVOVALVE R AD 832692
DC SHAH HYDRAULIC RESEARCH / MFG CO BURBANK CALIF
R NWC-TP-4519 0368 NAVAL WEAPONS CENTER CHINA LAKE CALIF

213 AIRCRAFT FLT CONT SYS FIELD SAFETY EXPERIENCE R AD 894814
BIELKA ET AL BOEING CO SEATTLE WASHINGTON
R AFFDL-TR-72-33 0572 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

214 TEST / ANAL OF QUADRUPLE REDUNDANT HORIZ STAB ACTUATING SYS R AD 901616
APPLEFORD ET AL BOEING CO SEATTLE WASHINGTON
R FAA-SS-72-70 0472 FEDERAL AVIATION ADMINISTRATION

215 LAB ANALYSIS OF SST DIRECT ELECT COMMAND / STAB SYS FAIL EFF R AD 901617
ML BEATTIE BOEING CO SEATTLE WASHINGTON
R FAA-SS-72-71 0772 FEDERAL AVIATION ADMINISTRATION

216 DIGITAL AIRBORNE DATA SYSTEM R AD 892890
JD ANDERSON ET AL HUGHES AIRCRAFT CO CULVER CITY CALIF
R AFFDL-TR-71-33 0471 AIR FORCE FLT DYNAMICS LAB WPAFB OHIO

APPENDIX C

LETTER SURVEY

OF

POTENTIAL SUPPLIERS, MANUFACTURERS AND EQUIPMENT USERS

Columbus Division
North American Rockwell

4300 East Fifth Avenue
Columbus, Ohio 43216

SAMPLE

October 23, 1973

Abex Corporation
Aerospace Division
3151 West Fifth Street
Oxnard, California 93030

Attention: Mr. Arthur Marshall, Product Sales Manager

Subject: Multi-Channel, Digital Fly-By-Wire Actuation Systems

Gentlemen:

Rockwell International, under contract with NASA, is conducting a study to determine the most suitable approach for development of an aircraft flight control actuation system for use in an advanced, all-digital, redundant, multi-channel, fly-by-wire system. A goal of this study is to select the most appropriate actuation and feedback sensor design approach which can meet the digital FBW system military and commercial requirements. The actuation system will be developed for flight demonstration in the NASA Phase II aircraft (F-8C) in the 1976 time period, but must have general applicability for use on future military and commercial aircraft which might utilize advanced, redundant, digital flight control concepts.

An initial task of this study is to survey, compile, review, and summarize the state-of-the-art of actuation devices which can be either directly controlled by digital signals or by intermediate D/A conversion.

Results of the survey will be compiled, published, and made available to the technical community. Due credit for all unique ideas, concepts, approaches, designs, etc. will of course be acknowledged. The survey is expected to be beneficial to everyone as a media for dissemination of ideas and information. It will be particularly beneficial to participants in providing an opportunity to extoll the virtues of each particular device, design, idea, approach, etc. and to receive published acknowledgement of your particular expertise. Although contractual funds are not available for supplier's effort under this study, a development contract for flight test hardware is anticipated during the follow-on phase. The follow-on phase will demonstrate the applicability of the digital fly-by-wire system to domestic and international transports, fixed wing business aircraft, and military tactical and transport aircraft.

This effort is being accomplished under the direction of Mr. Robert Averill, Technical Representative, NASA Langley Research Center.

Information regarding actuators, sensors, or actuation systems either produced or studied which may have application in a multi-channel digital FBW system would be very helpful and appreciated. An early reply (November 21, 1973) would not only be very helpful in the conduct of the survey but will assure due consideration in the published document.

Please submit all technical data and direct all inquiries and replies to Mr. Richard Hupp (phone 614-239-2713) or Mr. Robert Rossing (phone 614-239-2616).

The technical data should be forwarded to the following address:

Rockwell International Corporation
Columbus Aircraft Division
4300 East Fifth Avenue
Columbus, Ohio 43216

Attention: Mr. R. Hupp
Department 71

Your participation in the above survey of actuation and sensor devices suitable for use in primary flight controls is solicited.

Very truly yours,

ROCKWELL INTERNATIONAL CORPORATION
Columbus Aircraft Division

R A Fronk

R. A. Fronk, Manager
Equipment, Systems & Hardware Section
Purchasing Department

RAF:pcc

cc: R. Hupp

Abex Corporation
Aerospace Division
3151 West Fifth Street
Oxnard, California 93030

AiResearch Mfg. Company
9851 Sepulveda Blvd.
Los Angeles, California 90009

Allegheny Air Lines, Inc.
National Airport
Washington, D.C. 20001

American Airlines, Inc.
633 Third Avenue
New York, New York 10017

Ametek/Instruments and Controls
860 Pennsylvania Boulevard
Feasterville, Pennsylvania 19047

Arvin Systems Inc.
1771 Springfield St.
Dayton, Ohio 45403

Baldwin Electronics Inc.
1101 McAlmot St.
P.O. Box 3838
Little Rock, Arkansas 12203

Battelle Memorial Institute
Battelle-Columbus Laboratories Div.
505 King Avenue
Columbus, Ohio 43201

Beaver Precision Products, Inc.
Sub. of Warner Electric Brake &
Clutch Co.
1970 Bid Beaver Road
Troy, Michigan 48084

Bell Helicopter Company
P. O. Box 482
Fort Worth, Texas 76101

The Bendix Corporation
20800 Civic Center Drive
Southfield, Michigan 48076

The Bendix Corporation
Bendix Electrodynamics Division
1600 Sherman Way
North Hollywood, California 91605

Bendix Corporation
Navigation and Control Division
43 Williams Avenue
Teterboro, New Jersey 07608

Bertea Corporation
18001 Von Karman Avenue
Irvine, California 92664

The Boeing Company
3801 S. Oliver
Wichita, Kansas 67210

The Boeing Company
7755 E. Margnl Way
Seattle, Washington 98124

The Boeing Company
The Vertol Division
Boeing Center
P.O. Box 16858
Philadelphia, Pennsylvania 19142

Bourns Incorporated
Trimpot Division
1200 Columbia Avenue
Riverside, California 92507

Braniff Airways, Inc.
Braniff Tower
P. O. Box 35001
Exhange Park
Dallas, Texas 75235

Cadallic Gage Co.
25760 Grovesbeck Hwy.
Warren, Michigan 48090

Chandler Evans Control Systems Div.
Colt Industries
Charter Oak Blvd.
West Hartford, Conn. 06101

Columbia Research Laboratories, Inc.
McDale Boulevard and Bullens Lane
Woodlyn, Pennsylvania 19094

Continental Airlines, Inc.
International Airport
Los Angeles, California 90009

Cornell Aeronautical Lab., Inc.
P.O. Box 235, 4455 Genesee St.
Buffalo, New York 14221

Curtiss Wright Corporation
Caldwell Division
Fairfield, New Jersey 07006

Delco Products
Division of General Motors
2000 Forrer Boulevard
Dayton, Ohio 45401

Delta Air Lines, Inc.
Hartsfield-Atlanta International
Airport
Atlanta, Georgia 30320

Digital Equipment Corporation
146 Main Street
Maynard, Massachusetts 01754

Douglas Aircraft Co.
Div. of McDonnell Douglas Corp.
3855 Lakewood Blvd.
Long Beach, California 90801

E. Systems Inc.
Montek Div.
2208 South 3270 West
Salt Lake City, Utah 84119

Eastern Air Lines Inc.
10 Rockefeller Plaza
New York, New York 10020

Ellanef Manufacturing Corp.
97-11 50th Avenue
Corona, New York 11368

Ellanef Manufacturing Corp.
Columbus Engineering Office
Port Columbus, Ohio

Fairchild Republic Division
Fairchild Industries, Inc.
Farmingdale, Long Island, N.Y.
11735

Flight Research Center
Edwards, California

Frontier Airlines Inc.
8250 Smith Road
Denver, Colorado 80207

G&H Technology, Inc.
1649 17th Street
Santa Monica, California 90404

G. L. Collins Corporation
5875 Obisoqpo Avenue
Long Beach, California 90805

General Dynamics Corporation
Convair Aerospace Div.
P.O. Box 80877
San Diego, California 92138

General Electric Co.
Aerospace Instruments & Control
Systems
P.O. Box 5000
Binghampton, New York 13902

General Electric Company
Fluidics Operation
General Purpose Control Dept.
1 River Road
Schenectady, New York 12305

General Electric Company
West Coast Representative
P.O. Box 2830 Terminal Annex
Los Angles, California 90054

General Metals Corporation
Adel Aerospace Division
10777 Vanowen, P.O. Box 671
Burbank, California

General Metals Corporation
Adel Precision Division
36 Columbus Avenue
San Francisco, California 94111

Grumman Aerospace Corporation
S. Oyster Bay Road
Bethpage, Long Island, N.Y.11714

Gulton Industries Inc.
Instrumentation Products Div. - West
13030 Cerise
Hawthorne, California 90250

Gulton Industries Inc.
Servonic Instruments Division
1644 Whittier Avenue
Costa Mesa, California 92626

Hamilton Standard
Div. United Aircraft
Windsor Locks, Connecticut 06096

Hewlett-Packard Company
Automatic Measurement Division
395 Page Mill Road
Palo Alto, California 94304

HLM, Inc.
5 Harrison Avenue
Waldwrick, New Jersey 07463

Honeywell Inc.
Apparatus Control Division
2701 Fourth Avenue, South
Minneapolis, Minnesota 55408

Honeywell
Government & Aeronautical
Products Div.
2600 Ridgway Parkway
Minneapolis, Minnesota 55413

Hoover Electric Company
2100 S. Stoner Avenue
Los Angles, California 90025

Hydraulic Research & Manufacturing
Company
Division of Textron, Inc.
25200 W. Rye Canyon Road
Valencia, California 91355

IBM
Industrial Products
1271 Avenue of the Americas
New York, New York 10020

Icon Corporation
156 Sixth St.
Cambridge, Mass. 02142

IMC Magnetic Corporation
Eastern Division
570 Main Street
Westbury, New York 11591

Johnson Space Center
Huston, Texas

Kavlico Electronics Inc.
20869 Plummer Street
Chatsworth, California 91311

Kearfott Division
Singer-General Precision Inc.
1150 McBride Avenue
Little Falls, N.J. 07424

Lear Siegler Inc.
Astronics Divison
3171 W. Bundy Drive
Santa Monica, California 90406

Lear Siegler, Inc.
Power Equipment Division
17600 Broadway Avenue
Maple Heights, Ohio 44137

Litton Industries/Allis (Louis) Co.
427 E. Stewart St.
Milwaukee, Wisconsin 53201

Litton Precision Gear
Division of Litton Systems, Inc.
4545 Southwestern Blvd.
Chicago, Illinois 60609

Lockheed Aircraft Corporation
2555 N. Hollywood Way
Burbank, California 91503

Lockheed-Georgia Company
86 South Cobb Drive
Marietta, Georgia 30060

LTV Aeronautics Corporation
P.O. Box 5907
1525 Elm Street
Dallas, Texas 75222

M P C Products Corporation
4200 W. Victoria
Chicago, Illinois 60646

Marshall Space Flight Center
Huntsville, Alabama

Martin Marietta Aerospace
1800 K Street, N.W.
Washington, D.C. 20006

McDonnell Douglas Corporation
Lambert Airport
St. Louis, Missouri 63166

Moog, Inc.
Proner Airport
East Aurora, New York 14052

Nash Controls
1275 Bloomfield Avenue
Caldwell, New Jersey 07006

National Airlines Inc.
P.O. Box 2055
Airport Mail Facility
Miami, Florida 33159

National Waterlift Co.
Div. Pneumo Dynamics Corporation
2220 Palmer Avenue
Kalamazoo, Michigan 49001

North Central Airlines, Inc.
7500 Northliner Drive
Minneapolis, Minnesota 55450

Northrop Corporation
1800 Century Park East
Century City, California 90067

Northwest Airlines Inc.
Minneapolis-St. Paul International
Airport
St. Paul, Minnesota 55111

Ozark Air Lines Inc.
Box 10007
Lambert Field
St. Louis, Missouri 63145

Pacific Control
P.O. Box 7127
Burbank, California 91505

Pan American World Airways, Inc.
Pan Am Bldg.
New York, New York 10017

Parker Aircraft
18321 Jamboree Blvd.
Irvine, California 92664

The Perkin-Elmer Corporation
Electro-Optical Division
50 Danbury Road
Wilton, Connecticut 06897

Photomation Inc.
280 Polaris Avenue
Mountain View, California 94040

Pickering and Company, Inc.
101 Sunnyside Boulevard
Plainview, Long Island, N.Y. 11803

Plessey Industries, Inc.
Plessey Dynamics Division
1414 Chestnut Avenue
Hillside, New Jersey 07205

Process Systems Inc.
356 West Seventh Street, South
Salt Lake City, Utah 84101

Rockwell International Corporation
Autonetics Div., Headquarters
P.O. Box 4192
Anaheim, California

Rockwell International Corporation
B-1 Division Headquarters
5701 W. Imperial Highway
Los Angeles, California

Rockwell International Corp.
Space Division Headquarters
12214 Lakewood Blvd.
Downey, California 90241

Ronson Corporation
Woodbridge
New Jersey 07095

Ronson Hydraulic Unit
c/o Magna Engineering Sales Co., Inc.
7949 Keller Road
Cincinnati, Ohio 45243

Schaevitz Engineering Company
P.O. Box 505
Pennsauken, New Jersey 08101

Skurka Engineering Company
P.O. Box 32158-T
Los Angeles, California 90032

Southern Airways, Inc.
Atlanta Airport
Atlanta, Georgia 30320

Southwest Research Institute
8500 Culebra Road
San Antonio, Texas 78284

Spar Aerospace Products Limited
825 Caledonia Road
Toronto, 395, Ontario, Canada

Sperry Rand Corp.
Sperry Flight Systems Div.
21111 N. 19th Avenue
Phoenix, Arizona 85002

Sperry Rand Corporation
Vickers Aerospace - Ordnance
Marine Div. of Sperry Rand Corp.
Troy, Michigan 48084

Sundstrand Aircraft Equipment Div.
Sundstrand Aviation
4747 Harrison Avenue
Rockford, Illinois 61101

Tektronix Inc.
Box 500
Beaverton, Oregon 97005

Teledyne Gurley
514 Fulton St.
Troy, New York 12181

Trans World Airlines, Inc.
Administrative Center
11500 Ambassador Drive
Kansas City, Missouri

United Air Lines Inc.
P.O. Box 66100
Chicago, Illinois 60666

United Aircraft Corporation
Norden Division
Helen St.
Norwalk, Connecticut 06851

United Aircraft Corporation
Sikorsky Aircraft Division
North Main Street
Stratford, Connecticut 06497

United States Army Aviation
Systems Command
St. Louis, Missouri

Western Gear Corporation
Aerospace Group
P.O. Box 190
Lynwood, California 90262

Westinghouse Electric
General Control
Buffalo, New York 14240

Weston Hydraulics
Division of Borg-Warner Corp.
7500 Tyrone Avenue
Van Nuys, California 91409

Wright-Patterson AFB
Airforce Dynamics Laboratory
Wright-Patterson AFB
Wright-Patterson, Ohio

APPENDIX D

PLANT VISITS

VISITED AND VISITING COMPANIES

Personal visitations by Messrs. R. Rossing, and R. Hupp were made to the companies and agencies listed below:

October 22	-	Bertea Corp. & Cadallic Controls
23	-	Autonetics & Space Information Divisions of Rockwell International
24	-	Los Angeles Division of Rockwell International
25	-	AiResearch Mfg. Co.
26	-	Hydraulic Research & Mfg. Co. & Abex Corp.
29	-	Flight Research Center at Edwards AFB
30	-	Lear Siegler Inc. (Astronics Div.)
31	-	Johnson Space Center at Houston
November 1	-	Marshall Space Flight Center and Army Missile Command Facility at Huntsville

Mr. Robert Averill arranged and conducted the visitations with the NASA facilities.

Messrs. R. Rossing and R. Hupp continued the personal visitations initiated during the first month of the subject study with a visit to Wright Patterson Air Force Base on 6 November 1973. Mr. R. Rossing also visited the Bendix Research Laboratories, Southfield, Michigan, on 11 December 1973. Representa-tivies of Bendix Electrodynamics Division were also present. Visits were made to the Columbus facility to discuss the subject study by the following companies:

Lear Siegler, Inc. Maple Heights, Ohio	11-7-73
Sperry Flight Systems Phoenix, Arizona	11-13-73
Sperry Vickers Troy, Michigan	11-27-73
Sundstrand Aircraft Equipment Division Rockford, Illinois	11-28-73
Moog Incorporated East Aurora, New York	11-29-73
General Electric Co. Binghamton, New York	12-11-73
Sperry Flight Systems Phoenix, Arizona	12-14-73

APPENDIX E

RELIABILITY DATA

FAILURE RATES

SYSTEM ELEMENT	FAILURES/10^6 HRS	DATA SOURCE
HYD PRIMARY ACTUATOR		
Binding	.5	1
Jam	.08	1
Broken	2.7	1
Hyd Loss (Single System)	1.0	1
Maintenance Failure	600.0	1
HYD SECONDARY ACTUATORS		
Binding	.7	1
Jam	.07	1
Broken	.06	1
Hyd Loss (Single System)	.06	1
Maintenance Failure	64.0	1
HYD PUMPS	100	1
HYD MOTORS	2 to 100	1
HYD LINES (Hoses, Fitting, etc.)	.33/ft	1
HYD COMPONENTS (Misc.)	.2	1
HYD SYSTEM (Single Pumps)	140 (178)	1, 2
PM TORQUE MOTOR VALVE		
Catastrophic	0.02	Est.
Maintenance	5.0	Est.
TWO STAGE E/H VALVE	100.0 (40-2000)	4, 3
ONE STAGE E/H VALVE	60 (28)	4, 8
SPOOL/SLEEVE VALVE (Dual)		
Jam	0.2	Est.
Maintenance Failure	40.0	Est.
AMPLIFIER	7.	Est.
LVDT	4.0	6

FAILURE RATES (Continued)

SYSTEM ELEMENT	FAILURES/10^6 HRS	DATA SOURCE
DEMODULATOR	4	4
D/A CONVERSION	3	6
A/D CONVERSION	6	6
STEPPER MOTOR	25	4
MECHANICAL ACTUATORS (SECONDARY)		
Binding	1.0	1
Jam	.06	1
Break	.1	1
Hardover	1.8	1
Inoperative	1.7	1
GEAR BOXES	2-20	4
PUSH RODS		
Jam	.01	1
Break	.3	1
Binding	.04	1
CONTROL STICK ASSEMBLY		
Jam	.09	1
LINKAGE/BELLCRANK		
Jam	.09	1
Broken	.12	1
PULLEY/SECTORS		
Jam	.06	1
Broken	.07	1
MECHANICAL DISCONNECTS (Per Pivot)	.04	1
CONTROL CABLES	.01	5
BEARINGS	5.0	6
INDUCTION MOTORS	50.0	6

FAILURE RATES (Continued)

SYSTEM ELEMENT	FAILURES/10^6 HRS	DATA SOURCE
ELECTRICAL SYSTEM (Single)	128	7
ELECTRICAL SYSTEM (Dual)	14	5
ENGINE	30	5
CONTROL SURFACE	.01	5
SINGLE COMPUTER CHANNEL	200.	Est.

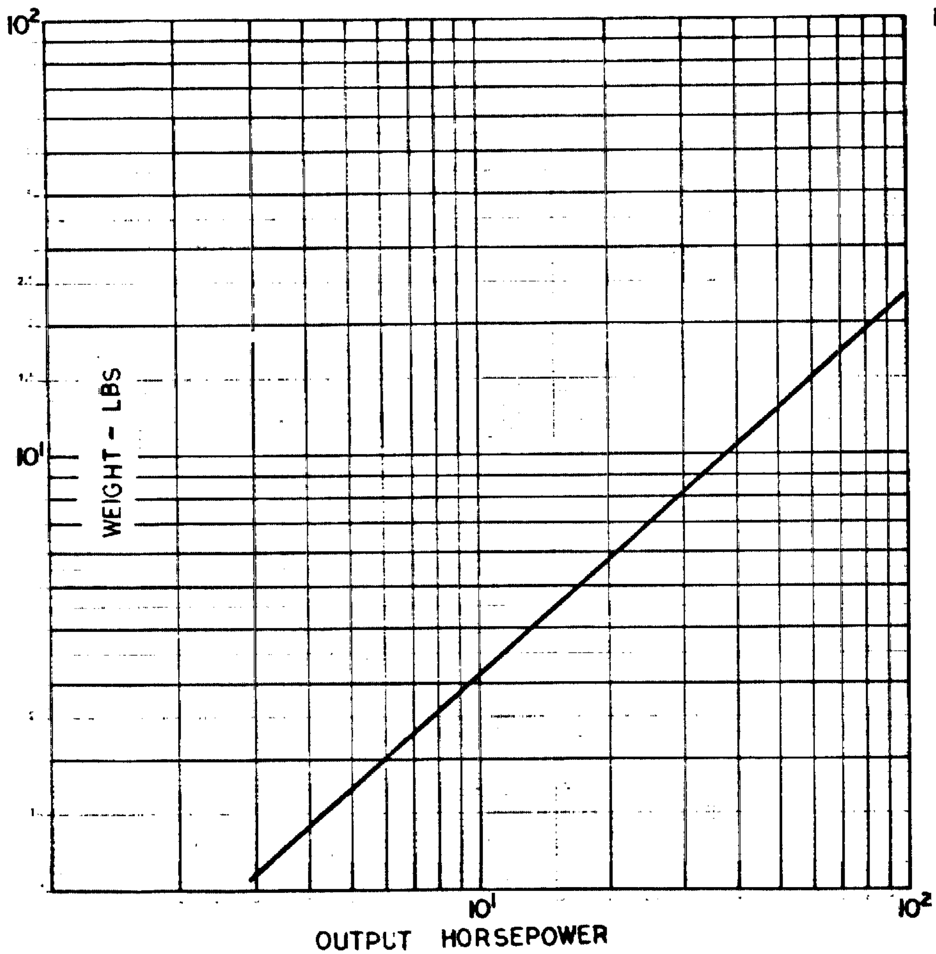
DATA SOURCES:

1. F-4, F-8 and A-7 TABS (CAD derived from Naval Safety Center Flt. Contr. Incident Reports - Sys. 2.3 x 10^6 Flt. Hr. Sample)
2. TR-68-1
3. TR-65-80
4. Failure Rate Data Handbook (FARADA) Bureau of Naval Weapons
5. TR-70-135
6. MIL-217A Reliability Stress and Failure Rate Data for Electronic Equipment
7. Reliability and Redundancy Study for Electronic Flight Control Systems (Honeywell Dc. No. 21718-FR)
8. Abex reported data on United Airlines 727 and 737 Experience (9.7 million flight hours)

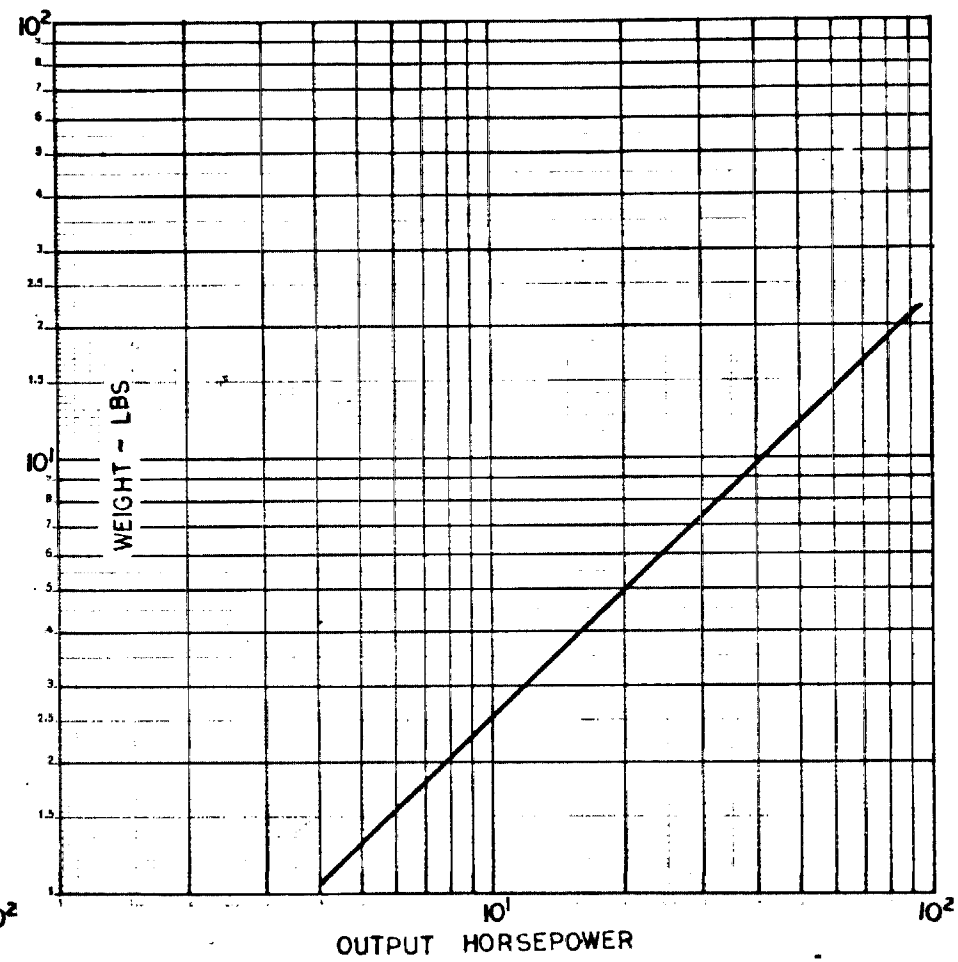
APPENDIX F

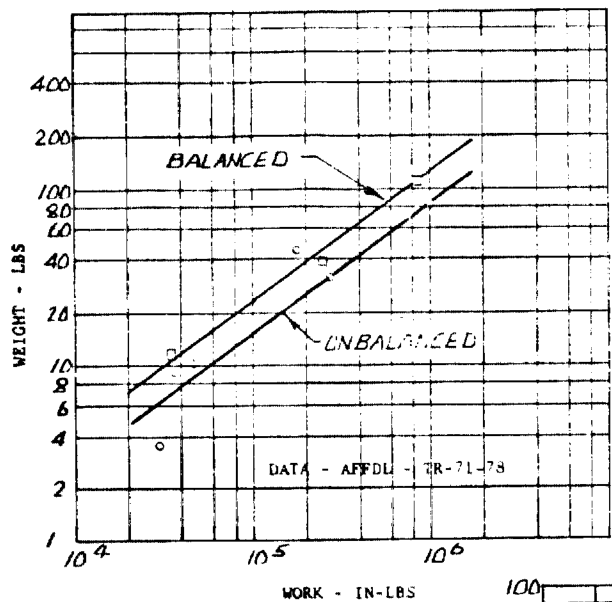
WEIGHT DATA

HYDRAULIC PUMPS

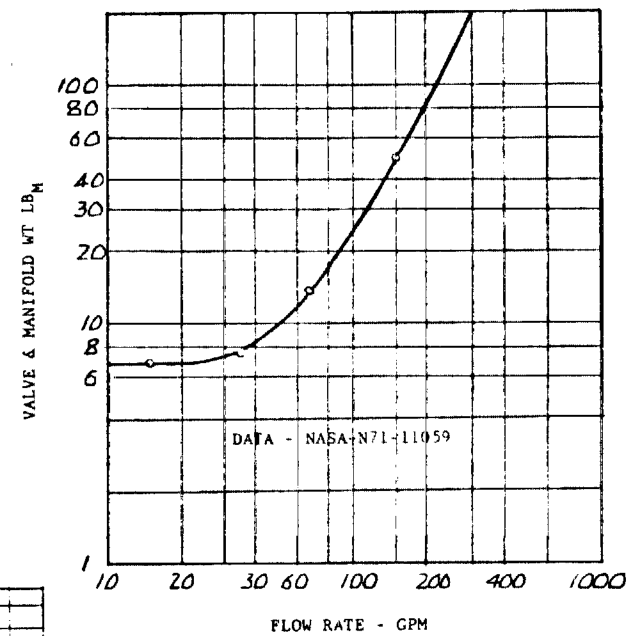


HYDRAULIC MOTORS

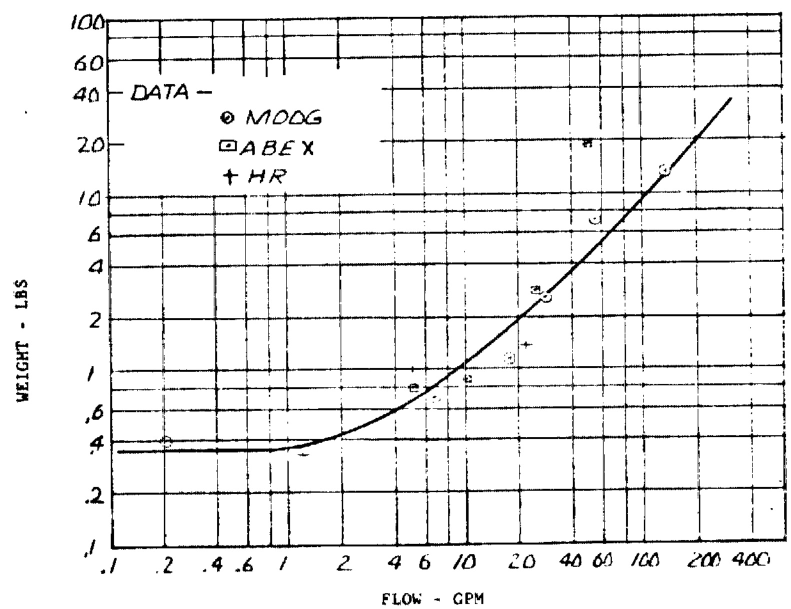




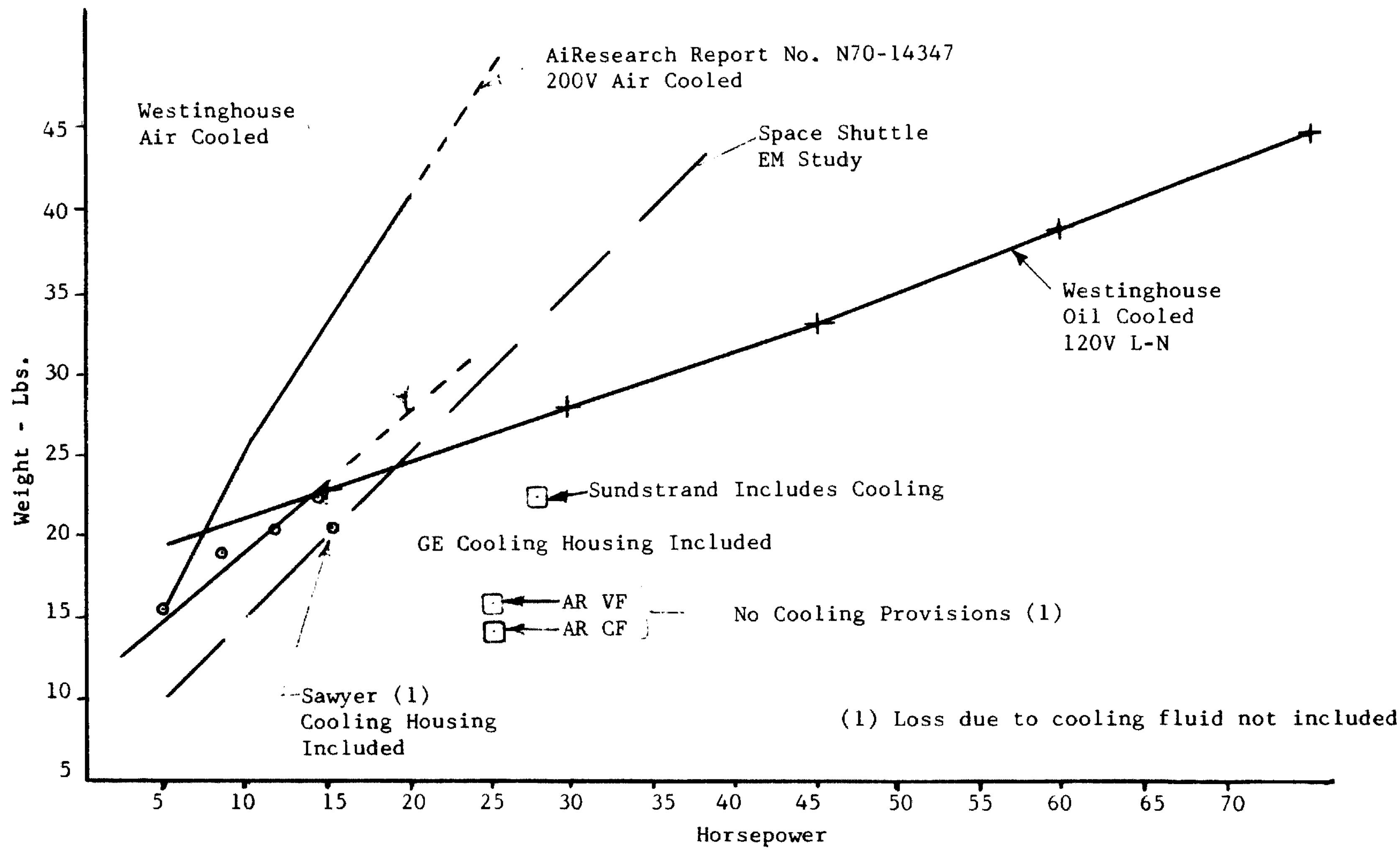
DUAL TANDEM
ACTUATORS



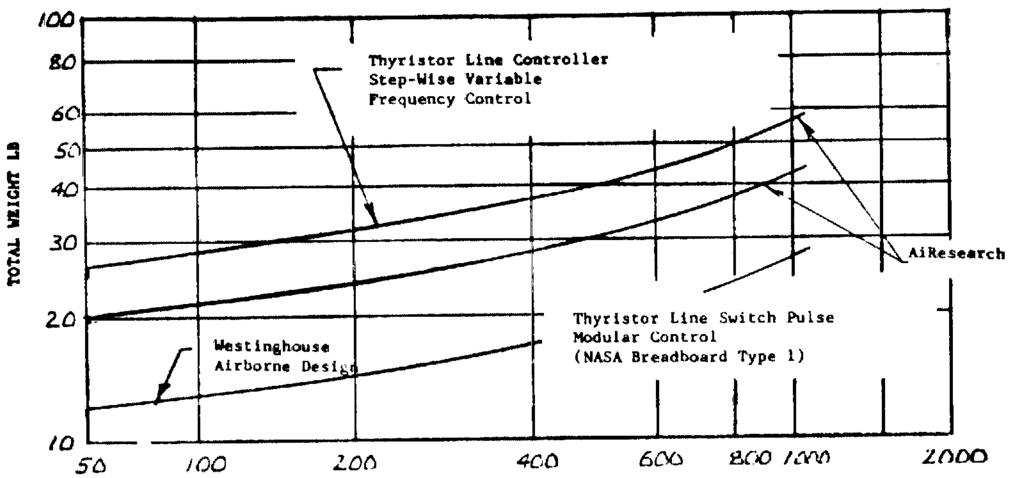
SPOOL-SLEEVE VALVES



ELECTROHYDRAULIC
SERVO VALVE WEIGHT

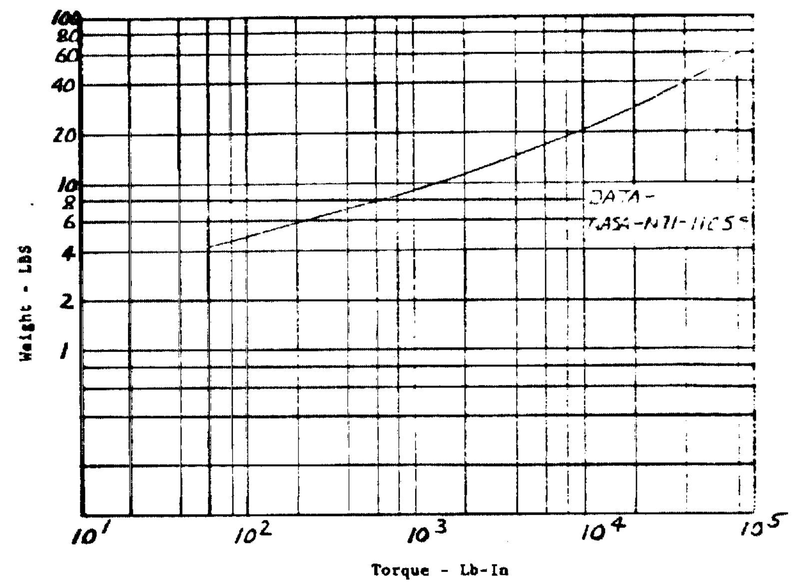


INDUCTION MOTORS (3Ø 4POLE)

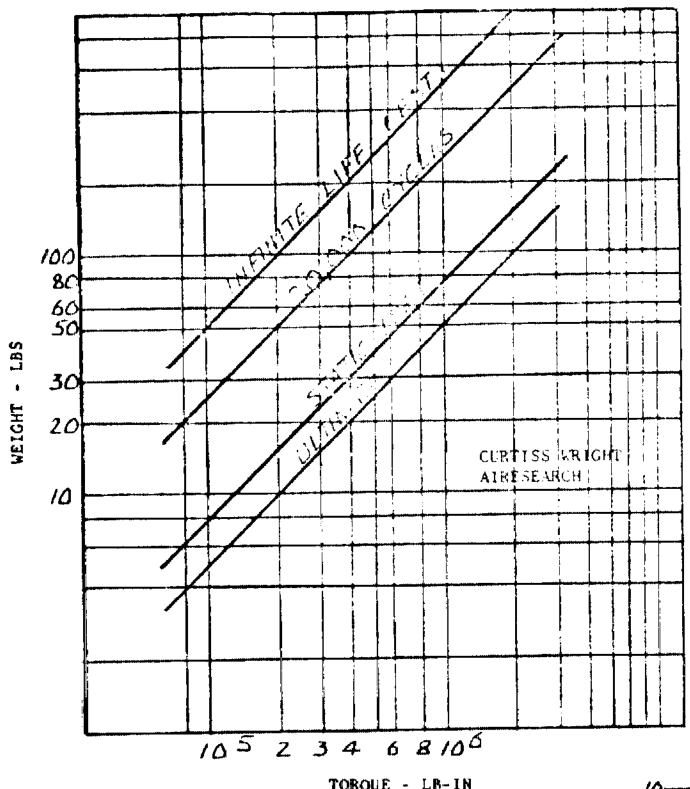


NOTES: •Based on 160°C case temperature - liquid cooled.
•5 modules for each motor provide reversing and all phase protection.
•Includes module control logic.

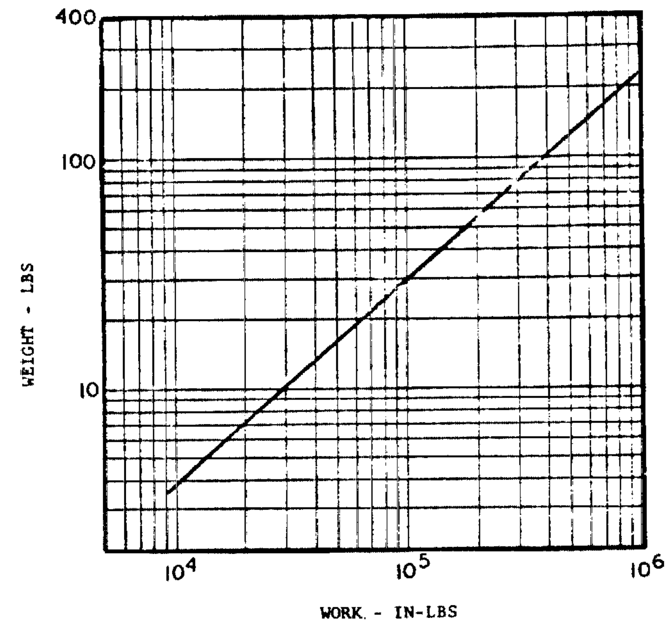
MOTOR CONTROLLER



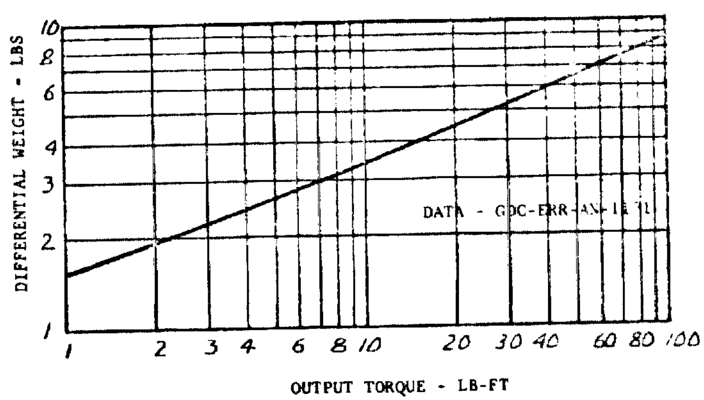
SPRING CLUTCH



MECHANICAL ROTARY ACTUATORS



BALLSCREW ACTUATORS



DIFFERENTIAL WEIGHT

APPENDIX G

RELIABILITY ANALYSIS

RELIABILITY ANALYSIS

There are two reliability figures of interest: operational reliability and maintenance reliability. The first reliability figure is a measure of safety; the second reliability figure is measure of the maintenance effort expectations. Maintenance reliability is simply one minus the sum of the component failure rates and is quite easily calculated. The operational reliability is a bit more complex. The operational reliability of the three approaches, the secondary actuator, the PM torque motor, and the stepper motor approaches, are calculated in the following paragraphs.

The system is considered operational when some measure of control is provided even though degraded in performance. The number of failures the system can withstand and still provide a measure of control is dependent upon which elements fail and in what manner. The system cannot withstand any failure of a series element. A four-channel system can provide some control after three open type failures in the parallel section. It cannot stand two hardover failures in the same direction since the two remaining channels are required to balance the hardover leaving nothing for control. To account for this phenomena, three failure states are considered: positive hardovers, negative hardovers, and opens. Each element is considered to be in one of the four states, the operational state or one of the three failure states. The parallel branches for the three and four channel systems are analyzed first and then combined with the series elements.

SECONDARY ACTUATOR

The reliability of the secondary actuator approach is depicted in Figure G-1. The diagram can be simplified by combining series elements. The probability of the series segments failing in a one-hour period, based upon the data contained in Appendix E and dual power supplies and amplifiers, are:

$$Q_A = (.001 + 3.0 + 3.0 + 4.0 + .001 + 60.0 + 1.0) \times 10^{-6}$$

$$= 71 \times 10^{-6}$$

$$Q_B = (.01 + .01 + .2 + .35 + .001 + .01) \times 10^{-6}$$

$$= 0.58 \times 10^{-6}$$

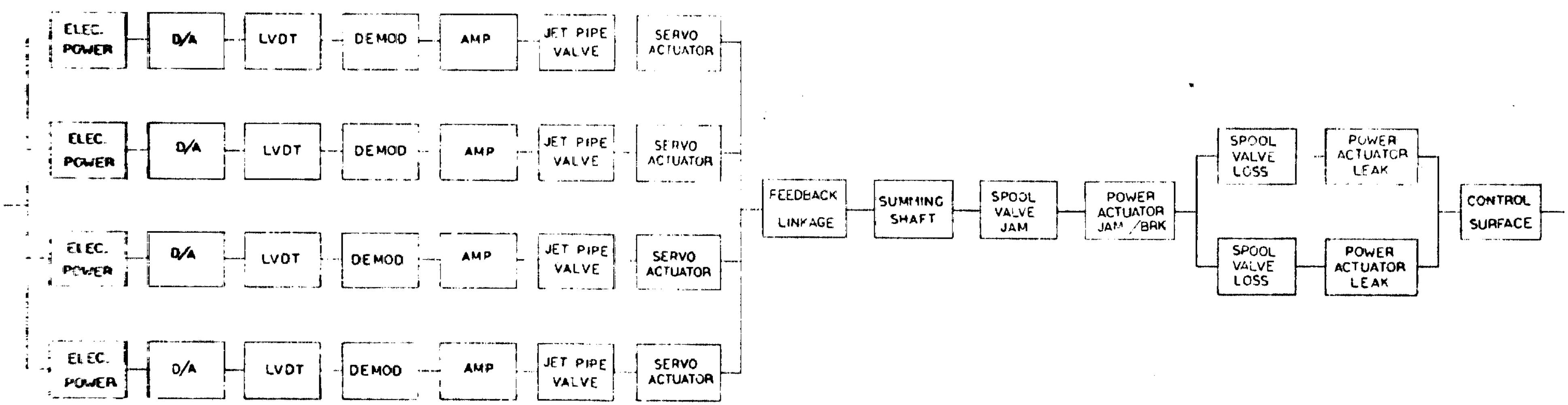


FIGURE G-1 SECONDARY ACTUATOR APPROACH RELIABILITY DIAGRAM

The simplified reliability diagram is shown in Figure G-2:

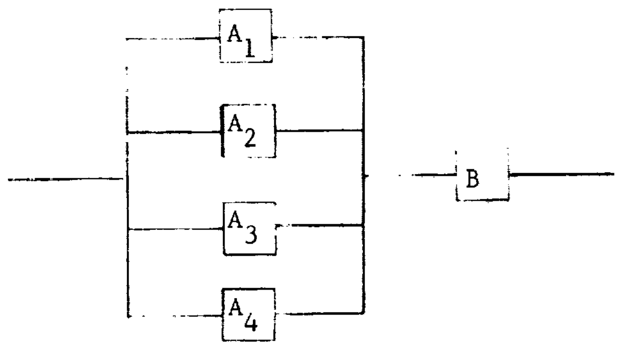


Figure G-2 Secondary Actuator Approach
Simplified Reliability Diagram

The probability of success (P(S)) is defined as being the event when some (any) level of control is provided for the entire mission period. The probability of success of the three and four channel systems are derived below, based upon the following:

- E_j - the event of i channels failing
- $H_i L_j F_k$ - the event of i channels failing hardover (+),
  - of j channels failing hardover (-),
  - and of k channels failing hardover open
- Q - probability of the failure event

$$Q \triangleq (q_H + q_L + q_F)\, Q$$

$$1 = P + Q$$

Considering the parallel segment first then:

<u>3 Channels</u>

$$
\begin{aligned}
P_A(S) &= P(S|E_0)\cdot P(E_0) \\
&\quad + P(S|E_1)\cdot P(E_1) \\
&\quad + P(S|E_2)\cdot P(E_2) \\
&\quad + P(S|E_3)\cdot P(E_3) \\
&= P(E_0) + P(E_1) + P(S|E_2)\cdot P(E_3)
\end{aligned}
$$

where $P(S|E_i)$ is the conditional probability of success given that the event E_i has occurred.

$$
\begin{aligned}
P(S|E_2) &= P(S|H_2L_0F_0|E_2)\cdot P(H_2L_0F_0|E_2) \\
&\quad + P(S|H_1L_1F_0|E_2)\cdot P(H_1L_1F_0|E_2) \\
&\quad + P(S|H_1L_0F_1|E_2)\cdot P(H_1L_0F_1|E_2) \\
&\quad + P(S|H_0L_2F_0|E_2)\cdot P(H_0L_2F_0|E_2) \\
&\quad + P(S|H_0L_1F_1|E_2)\cdot P(H_0L_1F_1|E_2) \\
&\quad + P(S|H_0L_0F_2|E_2)\cdot P(H_0L_0F_2|E_2) \\
&= P(H_1L_1F_0|E_2) + P(H_0L_0F_2|E_2) \\
&= \binom{2}{1} q_H q_L + q_F^2 \\
&= 2q_H q_L + q_F^2
\end{aligned}
$$

$$
\begin{aligned}
P(E_0) &= (1 - Q)^3 \qquad = 1 - 3Q + 3Q^2 - Q^3 \\
P(E_1) &= \binom{3}{1}(1 - Q)^2\, Q = 3Q(1 - Q)^2 \\
P(E_2) &= \binom{3}{2}(1 - Q)\, Q^2 = 3Q^2(1 - Q)
\end{aligned}
$$

Summation of terms produce:

$$P_A(S) = 1 - 3\left[1 - P(S|E_2)\right] Q^2 + 2\left[1 - \frac{3}{2} P(S|E_2)\right] Q^3$$

$$= 1 - 3Q^2\left[1 - (2q_H q_L + q_F^2)\right] + 2Q^3\left[1 - \frac{3}{2}(2q_H q_L + q_F^2)\right]$$

<u>4 Channels</u>

$$\begin{aligned} P_A(S) = \; & P(S|E_0) \cdot P(E_0) \\ & + P(S|E_1) \cdot P(E_1) \\ & + P(S|E_2) \cdot P(E_2) \\ & + P(S|E_3) \cdot P(E_3) \\ & + P(S|E_4) \cdot P(E_4) \end{aligned}$$

$$\begin{aligned} = \; & P(E_0) \\ & + P(E_1) \\ & + P(S|E_2) \cdot P(E_2) \\ & + P(S|E_3) \cdot P(E_3) \end{aligned}$$

$$\begin{aligned} P_A(S|E_2) = \; & P(S|H_2L_0F_0|E_2) \cdot P(H_2L_0F_0\ E_2) \\ & + P(S|H_1L_1F_0|E_2) \cdot P(H_1L_1F_0\ E_2) \\ & + P(S|H_1L_0F_1|E_2) \cdot P(H_1L_0F_1\ E_2) \\ & + P(S|H_0L_2F_0|E_2) \cdot P(H_0L_2F_0\ E_2) \\ & + P(S|H_0L_1F_1|E_2) \cdot P(H_0L_1F_1\ E_2) \\ & + P(S|H_0L_0F_2|E_2) \cdot P(H_0L_0F_2\ E_2) \end{aligned}$$

$$
\begin{aligned}
&= P(H_1L_1F_0|E_2) \\
&\quad +P(H_1L_0F_1|E_2) \\
&\quad +P(H_0L_1F_1|E_2) \\
&\quad +P(H_0L_0F_2|E_2)
\end{aligned}
$$

$$= \binom{2}{1}\ q_Hq_L = \binom{2}{1}\ q_Hq_F + \binom{2}{1}\ q_Lq_F + q_F^2$$

$$= 2\ q_Hq_L = 2\ q_Hq_F + 2\ q_Lq_F + q_F^2$$

$$
\begin{aligned}
P_A(S|E_3) \quad = \quad & P(S|H_3L_0F_0|E_3) \cdot P(H_3L_0F_0|E_3) \\
& +P(S|H_2L_1F_0|E_3) \cdot P(H_2L_1F_0|E_3) \\
& +P(S|H_2L_0F_1|E_3) \cdot P(H_2L_0F_1|E_3) \\
& +P(S|H_1L_2F_0|E_3) \cdot P(H_1L_2F_0|E_3) \\
& +P(S|H_1L_1F_1|E_3) \cdot P(H_1L_1F_1|E_3) \\
& +P(S|H_1L_0F_2|E_3) \cdot P(H_1L_0F_2|E_3) \\
& +P(S|H_0L_3F_0|E_3) \cdot P(H_0L_3F_0|E_3) \\
& +P(S|H_0L_2F_1|E_3) \cdot P(H_0L_2F_1|E_3) \\
& +P(S|H_0L_1F_2|E_3) \cdot P(H_0L_1F_2|E_3) \\
& +P(S|H_0L_0F_3|E_3) \cdot P(H_0L_0F_3|E_3)
\end{aligned}
$$

$$= P(H_1L_1F_1|E_3) + P(H_0L_0F_3|E_3)$$

$$= \binom{3}{1}\binom{2}{1}\ q_Hq_Lq_F + q_F^3$$

$$= 6\ q_Hq_Lq_F + q_F^3$$

$$P(E_0) = (1 - Q)^4 = 1 - 4Q + 6Q^2 - 4Q^3 + Q^4$$

$$P(E_1) = \binom{4}{1} (1 - Q)^3 Q = 4Q - 12 Q^2 + 12 Q^3 - 4 Q^4$$

$$P(E_2) = \binom{4}{2} (1 - Q)^2 Q^2 = 6Q^2 - 12Q^3 + 6Q^4$$

$$P(E_3) = \binom{4}{3} (1 - Q) Q^3 = 4Q^3 - 4Q^4$$

Summation of terms produce:

$$P_A(S) = 1 - 6Q^2 \left[1 - P(S|E_2)\right] - 4Q^3 \left[1 - P(S|E_3)\right] + 12Q^3 \left[1 - P(S|E_2)\right]$$

$$+ 2Q^4 \left[3P(S|E_2) - 2P(S|E_3)\right] - 3Q^4$$

If it is assumed that:

$$Q\text{'s} < .001$$

$$q_H = q_L = q_F = 1/3$$

then the probability of success through the parallel segment for the three and four channel systems are:

<u>3 Channels</u>

$$P_A(S) \approx 1 - 3Q^2 \left[1 - P(S|E_2)\right]$$

$$\approx 1 - 2Q^2$$

4 Channels

$$P_A(S) \approx 1 - 6Q^2 \left[1 - P(S|E_2)\right]$$

$$\approx 1 - \frac{4}{3} Q^2$$

Combining the parallel branch segment (A elements) with the series element segment (B elements) produces:

3 Channels

$$P(S) = P_A(S) \cdot P_B(S)$$

$$\approx \left[1 - 2Q_A^2\right] \left[1 - Q_B\right]$$

$$\approx 1 - Q_B - 2Q_A^2$$

4 Channels

$$P(S) = P_A(S) \; P_B(S)$$

$$\sim \left[1 - \frac{4}{3} Q_A^2\right] \left[1 - Q_B\right]$$

$$\approx 1 - Q_B - \frac{4}{3} Q_A^2$$

The four channel system is slightly more reliable than the three channel system, the difference being $\frac{2}{3} Q_A^2$. Both systems are primarily dependent upon the series elements, basically the actuator and power valve.

SYSTEM WITH MONITORS

The operational reliability of the basic system depicted in Figure G-2 with a monitor is calculated below. The monitor, by definition, can sense and disconnect failed parallel segment (A's). The monitor ceases to function when only two channels remain. The three and four channel reliability figures are:

3 Channels

$$P_A(S) = P(S|E_0)\cdot P(E_0)$$
$$+P(S|E_1)\cdot P(E_1)$$
$$+P(S|E_2)\cdot P(E_2)$$
$$+P(S|E_3)\cdot P(E_3)$$
$$= P(E_0) + P(E_1) + P(S|E_2)\cdot P(E_2)$$

$$P(S|E_2) = P(S|H_2L_0F_0|E_2) \cdot P(H_2L_0F_0|E_2)$$
$$+P(S|H_1L_1F_0|E_2) \cdot P(H_1L_1F_0|E_2)$$
$$+P(S|H_1L_0F_1|E_2) \cdot P(H_1L_0F_1|E_2)$$
$$+P(S|H_0L_2F_0|E_2) \cdot P(H_0L_2F_0|E_2)$$
$$+P(S|H_0L_1F_1|E_2) \cdot P(H_0L_1F_1|E_2)$$
$$+P(S|H_0L_0F_2|E_2) \cdot P(H_0L_0F_2|E_2)$$

$P(S|H_1L_0F_1|E_2)$ and $P(S|H_0L_1F_1|E_2)$ both are equal to ½ since they depend upon which failure occurs first and is removed by the monitor; if the hardover occurs first and is removed the system is operational; if the open occurs first the system is not operational. Both are equally likely to occur, therefore, the conditional probabilities are equal to ½. Therefore:

$$P(S|E_2) = \tfrac{1}{2}P(H_1L_0F_1|E_2) + \tfrac{1}{2}P(H_0L_1F_1|E_2) + P(H_0L_0F_2|E_2)$$
$$= \tfrac{1}{2}\binom{2}{1} q_H q_F + \tfrac{1}{2}\binom{2}{1} q_L q_F + q_F^2$$

$$P(E_0) = (1 - Q)^3$$
$$P(E_1) = \binom{3}{1} Q (1-Q)^2$$
$$P(E_2) = \binom{3}{2} Q^2 (1-Q)$$

Summing terms produces:

$$P_A = 1 - 3\left[1 - P(S|E_2)\right] Q^2 + \left[2 - 3P(S|E_2)\right] Q^3$$

4 Channels

$$\begin{aligned}
P_A(S) = {} & P(S|E_0) \cdot P(E_0) \\
& + P(S|E_1) \cdot P(E_1) \\
& + P(S|E_2) \cdot P(E_2) \\
& + P(S|E_3) \cdot P(E_3) \\
& + P(S|E_4) \cdot P(E_4) \\
= {} & P(E_0) + P(E_1) + P(E_2) + P(S|E_3) \cdot P(E_3)
\end{aligned}$$

$$\begin{aligned}
P(S|E_3) = {} & P(S|H_3L_0F_0|E_3) \cdot P(H_3L_0F_0|E_3) \\
& + P(S|H_2L_1F_0|E_3) \cdot P(H_2L_1F_0|E_3) \\
& + P(S|H_2L_0F_1|E_3) \cdot P(H_2L_0F_1|E_3) \\
& + P(S|H_1L_2F_0|E_3) \cdot P(H_1L_2F_0|E_3) \\
& + P(S|H_1L_1F_1|E_3) \cdot P(H_1L_1F_1|E_3) \\
& + P(S|H_1L_0F_2|E_3) \cdot P(H_1L_0F_2|E_3) \\
& + P(S|H_0L_3F_0|E_3) \cdot P(H_0L_3F_0|E_3) \\
& + P(S|H_0L_2F_1|E_3) \cdot P(H_0L_2F_1|E_3) \\
& + P(S|H_0L_1F_2|E_3) \cdot P(H_0L_1F_2|E_3) \\
& + P(S|H_0L_0F_3|E_3) \cdot P(H_0L_0F_3|E_3)
\end{aligned}$$

$$= \frac{1}{3} P(H_2L_0F_1|E_3) + \frac{1}{3} P(H_1L_1F_1|E_3) + \frac{2}{3} P(H_1L_0F_2|E_3)$$

$$+ \frac{1}{3} P(H_0L_2F_1|E_3) + \frac{2}{3} P(H_0L_1F_2|E_3) + P(H_0L_0F_3|E_3)$$

$$= \frac{1}{3} \binom{3}{1} q_H^2 q_F + \frac{1}{3} \binom{3}{1} \binom{2}{1} q_H q_L q_F + \frac{2}{3} \binom{3}{1} q_H q_F^2$$

$$+ \frac{1}{3} \binom{3}{1} q_L q_F^2 + \frac{2}{3} \binom{3}{1} q_L q_F^2 + q_F^3$$

$$P(E_0) = (1 - Q)^4$$

$$P(E_1) = \binom{4}{1} Q_A (1 - Q)^3$$

$$P(E_2) = \binom{4}{2} Q_A^2 (1 - Q)^2$$

$$P(E_3) = \binom{4}{3} Q_A^3 (1 - Q)$$

Summing terms produce:

$$P_A(S) = 1 - 4 \left[1 - P(S|E_3)\right] Q^3 + \left[3 - 4P(S|E_3)\right] Q^4$$

Assuming q's = 1/3 then the reliability of the three and four channel systems with monitoring is:

$$P_A(S) \approx 1 - Q_B - 2Q_A^2 \qquad @ \quad 3 \text{ channels}$$

$$P_A(S) \approx 1 - Q_B - \frac{8}{3} Q_A^3 \qquad @ \quad 4 \text{ channels}$$

Note that a monitor under these assumptions improves a four channel system operational reliability but does not improve a three channel system operational reliability. It could serve the purpose of reducing the performance degradation resulting from a failure.

Considering two states (operate and open failures) and defining the success event as being when at least two of the four A elements and the B element are operating the probability of success for the 3 and 4 channel systems without monitoring is:

3 Channels

$$P(S) = \left[P_A^3 + \binom{3}{2} P_A^2 Q_A \right] P_B$$

$$= 1 - Q_B - 3Q_A^2 (1 - Q_B) + 2Q_A^3 (1 - Q_B)$$

$$\approx 1 - Q_B - 3Q_A^2$$

4 Channels

$$P(S) = \left\{ 1 - \left[Q_A^4 + \binom{4}{3} P_A Q_A^3 \right] \right\} P_B$$

$$= \left\{ 1 - 4Q_A^3 + 3Q_A^4 \right\} (1 - Q_B)$$

$$= 1 - 4Q_A^3 + 3Q_B^4 - Q_B + Q_A^3 Q_B (4 + 3Q_A)$$

Assuming $Q_A < .001$ & $Q_B < .001$ then

$$P(S) \approx 1 - Q_B - 4Q_A^3$$

PM TORQUE MOTOR APPROACH

The reliability diagram of the PM torque motor approach is depicted in Figure G-3. The diagram can be simplified by combining series elements. The probability of the series segments failing in a one hour period, based upon Appendix E, are:

$$Q_A = (3.0 + 3.0 + 4.0) \times 10^{-6} = 10 \times 10^{-6}$$

$$Q_B = (9.0 + 21.0 + 2.0) \times 10^{-6} = 32 \times 10^{-6}$$

$$Q_C = (.01 + .20 + .35 + .001 + .01) \times 10^{-6} = 0.571 \times 10^{-6}$$

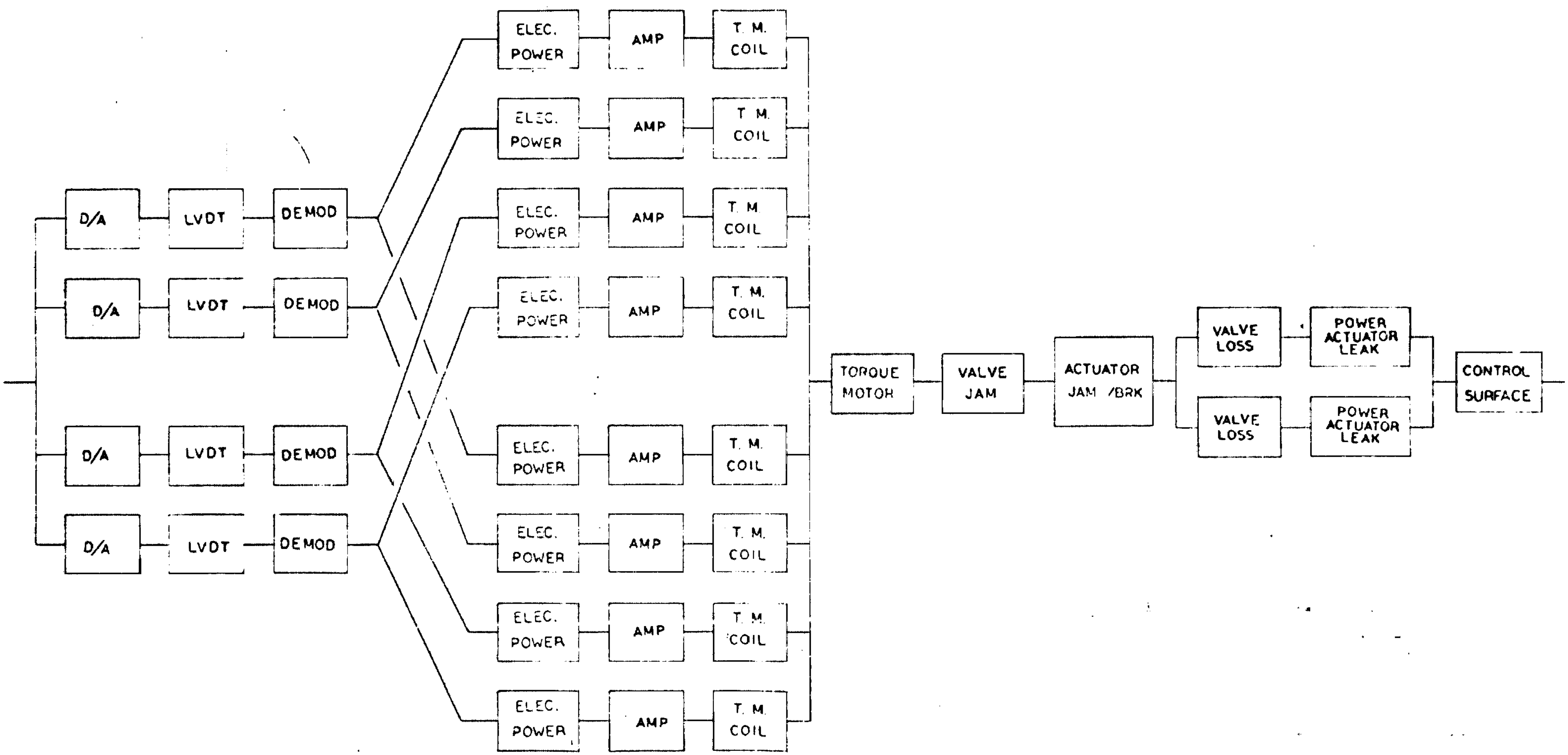
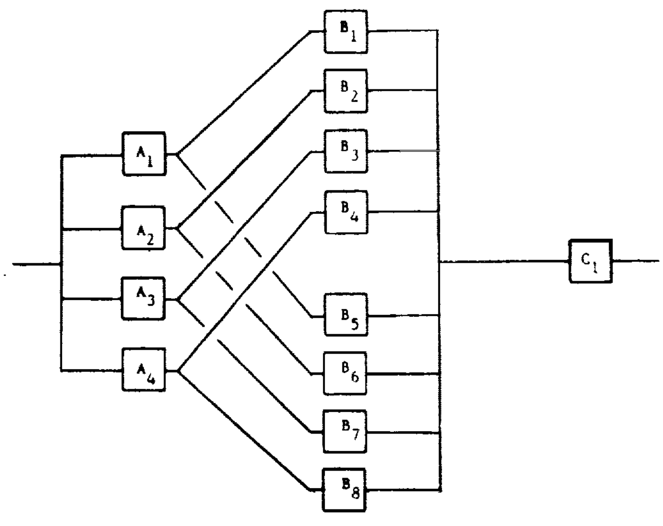
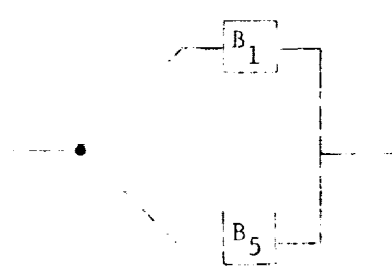


FIGURE G-3 PM TORQUE MOTOR DIRECT DRIVE CONCEPT

The simplified diagram is shown below:



The valve drive amplifiers are in parallel. Considering one channel through the parallel branch, the probability of a hardover is:



$$P(H) = P(H|E_0)\ P(E_0) + P(H|E_1)\ P(E_1) + P(H|E_2)\ P(E_2)$$

$$= P(H|E_2)\ P(E_2)$$

$$P(H|E_2) = P(H|H_2L_0F_0|E_2)\ P(H_2L_0F_0|E_2)$$

$$= +P(H|H_1L_1F_0|E_2)\ P(H_1L_1F_0|E_2)$$

$$= +P(H|H_1L_0F_1|E_2)\ P(H_1L_0F_1|E_2)$$

$$= +P(H|H_0L_2F_0|E_2)\ P(H_0L_2F_0|E_2)$$

$$= +P(H|H_0L_1F_1|E_2)\ P(H_0L_1F_1|E_2)$$

$$= +P(H|H_0L_0F_2|E_2)\ P(H_1L_0F_1|E_2)$$

$$= \left[P(H_2L_0F_0|E_2) + P(H_1L_0F_1|E_2)\right] P(E_2)$$

$$P(L) = P(L|E_2)\ P(E_2)$$

$$= \left[P(H_0L_2F_0) + P(H_0L_1F_1|E_2)\right]\ P(E_2)$$

$$P(F) = P(F|E_2)\ P(E_2)$$

$$= \left[P(H_1L_1F_0|E_2) + P(H_0L_0F_2|E_2)\right]\ P(E_2)$$

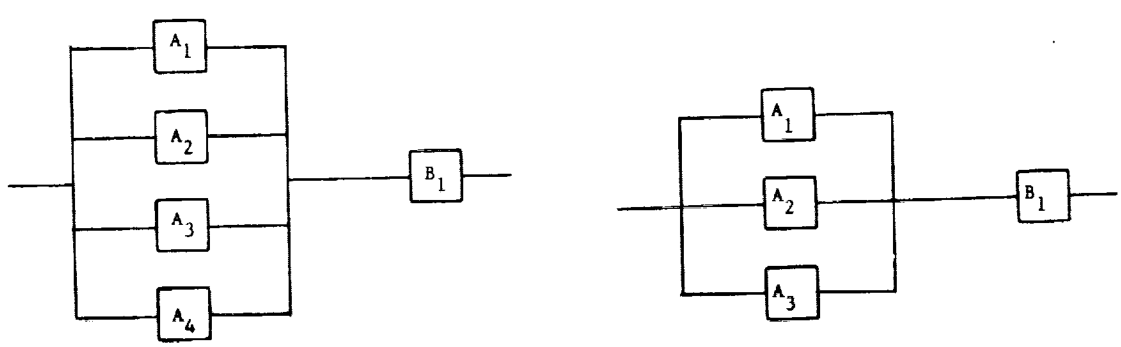
Assuming equally probability events, q's = 1/3 the three failure state probabilities are:

$$P(H) = \left[(\tfrac{1}{3})^3 + \binom{2}{1} (\tfrac{1}{3})^2 \right] P(E_2) = \tfrac{1}{3} P(E_2) = \tfrac{1}{3} Q^2$$

$$P(L) = \tfrac{1}{3} Q^2$$

$$P(F) = \tfrac{1}{3} Q^2$$

The relative distribution is unchanged (q's = 1/3). The probability of an operational type failure is reduced to Q^2. The reliability diagram can be further simplified by combing the parallel 'B' elements as above and then combining with the 'A' elements. With these changes, the PM torque motor approach reliability diagram reduces to a form equivalent to the simplified secondary actuation approach, Figure G-2, which has been analyzed previously. The simplified diagram is repeated below:



4 CHANNELS

3 CHANNELS

Failure probabilities for the above form are as follows:

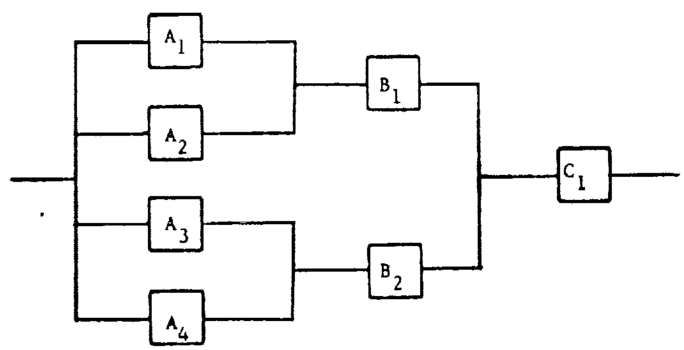
$$Q_A = (3.0 + 3.0 + 4.0) \times 10^{-6} = 10 \times 10^{-6}$$

$$Q_B = (32 \times 10^{-6})^2 + (.01 + .20 + .35 + .001 + .01) \times 10^{-6}$$

$$\approx 0.57 \times 10^{-6}$$

STEPPER MOTOR DIRECT DRIVE

The stepper motor direct drive approach is shown in Figure G-4. The diagram can be simplified by combining the series segment elements as depicted in the figure below.



The failure probabilities for the elements in the simplified diagram, based upon Appendix E, are:

$$Q_A = (35.0 + 79.0) \times 10^{-6} = 114 \times 10^{-6}$$

$$Q_B = (24 + .06) \times 10^{-6} = 25.06 \times 10^{-6}$$

$$Q_C = (.03 + .01 + .02 + .2 + .35 + .001 + .01) \times 10^{-6} = .62 \times 10^{-6}$$

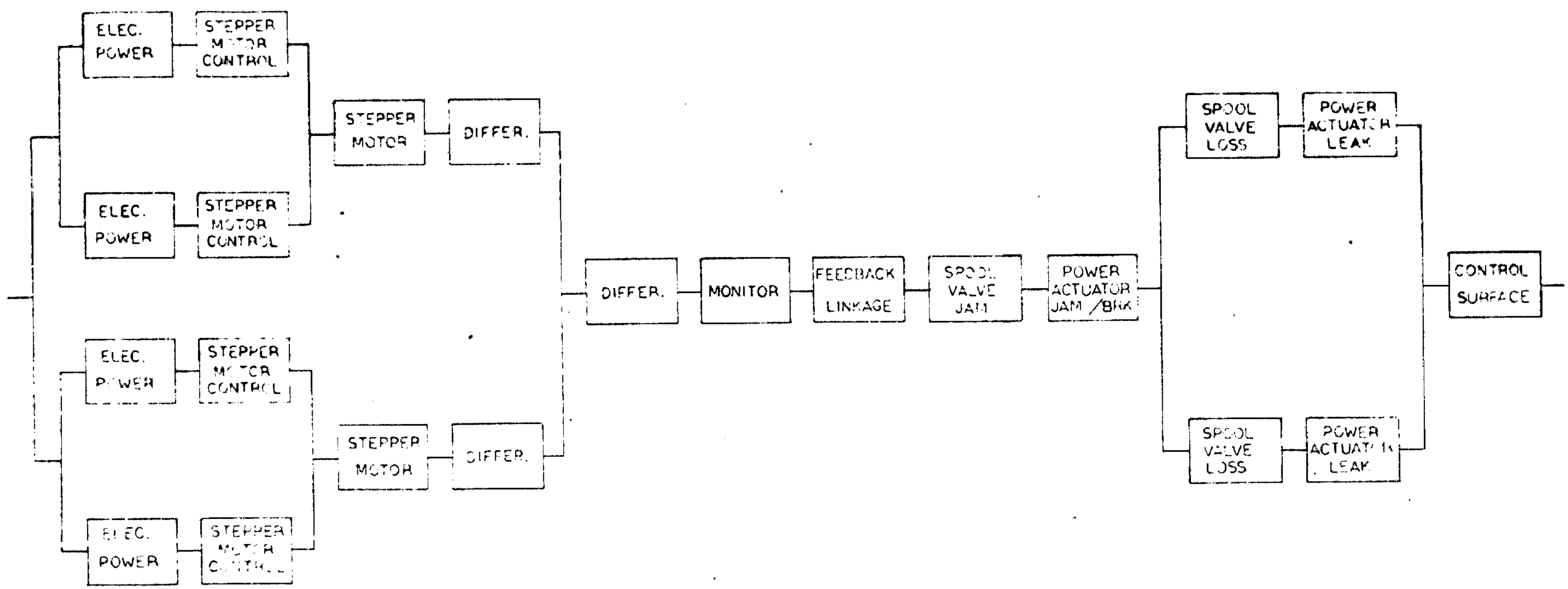


FIGURE G-4 DIRECT DRIVE STEPPER MOTOR

This approach is heavily dependent upon the monitor. For purposes of analysis, the monitor is assumed to prevent hardovers from entering element 'C'. The hardover potential is reflected in the monitor element which is placed in the series branch. The probability of the system being operational up to point 1 is:

$$P = \left[1 - Q_A^2\right]\left[1 - Q_B\right] = 1 - \left[Q_A^2 + Q_B - Q_A^2 Q_B\right]$$

$$\approx 1 - \left[Q_A^2 + Q_B\right]$$

The probability of failing is:

$$Q = Q_A^2 + Q_B$$

The probability of the system being operational is:

$$P = \left[1 - (Q_A^2 + Q_B)^2\right.\ \left[1 - Q_C\right]$$

$$\approx 1 - Q_B^2 - Q_C$$

The stepper motor approach is primarily dependent upon the series elements as is the secondary actuation and PM torque motor approaches; it is expected to be less reliable due to the additional series elements.

MAINTENANCE RELIABILITY

Maintenance reliability (MTBF) for the three approaches are reflected in the failure rates presented in Table G-1. These estimated failure rates are based upon data presented in Appendix E.

TABLE G-1 MAINTENANCE FAILURE RATE (FAILURES/10^6 HRS)

SECONDARY ACTUATOR APPROACH		PM TORQUE MOTOR APPROACH		STEPPER MOTOR APPROACH	
D/A (4)	12	D/A (4)	12	D/A (4)	12
Demod (4)	16	Demod (4)	16	Demod (4)	16
E/H Valves (4)	200	LVDT (Quad)	16	LVDT (Quad)	16
Secondary Actuator (4)	260	Amplifier (8)	256	Electr. (4)	316
LVDT (Quad)	16	Power Supplies (8)	70	Power Supply (4)	140
Amplifier (4)	120	Torque Motor (2)	10	Stepper Motor (2)	50
Power Supplies (8)	70	Linkage (1)	-	Linkage (1)	-
Summing Shaft (1)	-	Power/Valve (2)	40	Differential (3)	12
Linkage (1)	-	Power Actuator (1)	510	Ballscrew (1)	3
Power/Valve (2)	40			Power/Valve (2)	40
Power Actuator (1)	510			Actuator (1)	510
TOTAL	1244	TOTAL	930	TOTAL	1115